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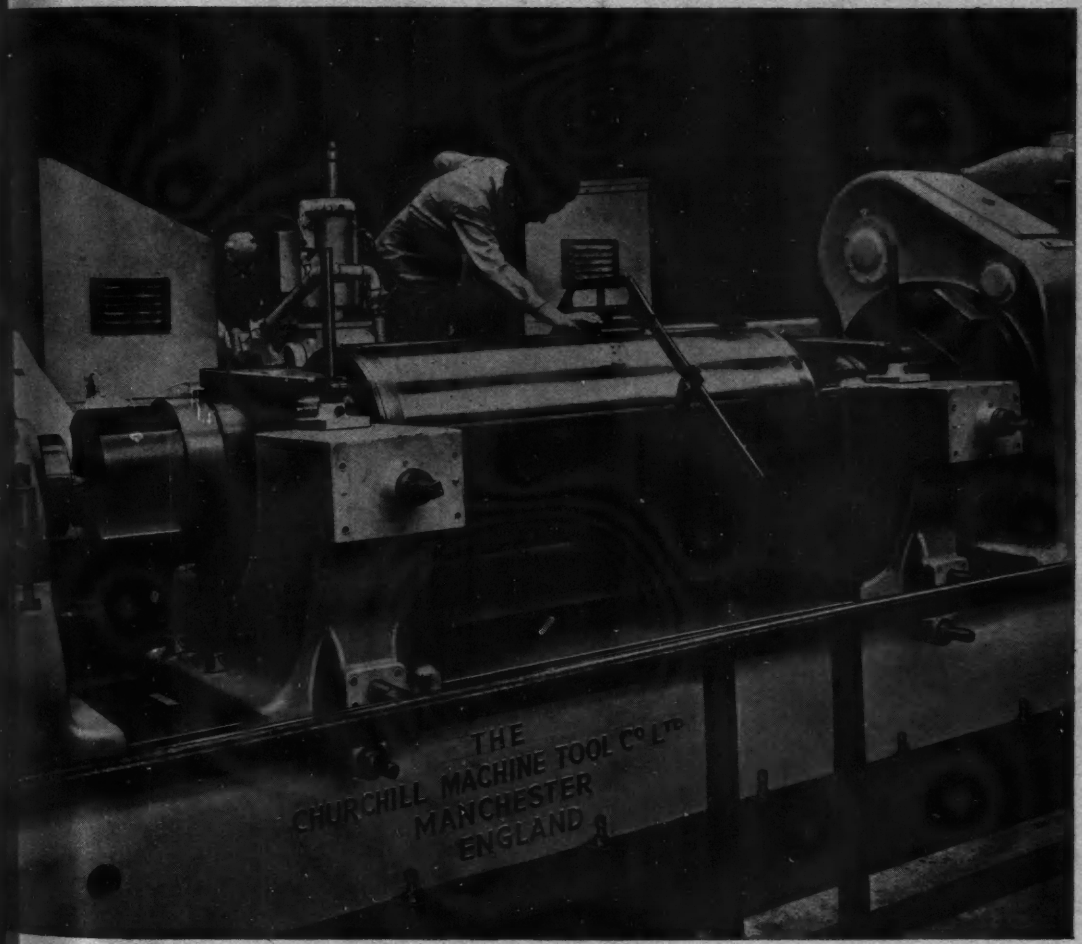

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FATIGUE STRENGTH OF CRANKSHAFTS OF LARGE DIESELS.

By Dr.-Ing. habil. E. LEHR and Dipl.-Ing. F. RUEF. (From *MTZ Motortechnische Zeitschrift*, Vol. 5, No. 11/12, December, 1943, pp. 349–357.)

OPERATIONAL reliability of a large diesel must largely depend upon a sufficiently high fatigue strength of the crankshaft. Up to the present, however, only the torsional and alternating bending fatigue strength of small crankshafts for automobile engines and aero-engines have been investigated, and no experimental results on the fatigue strength of the crankshafts of large diesels are available. Nor is it possible to judge large crankshafts with crankpins of, e.g., 250 mm. dia. on the basis of test data obtained with shafts of, say, 100 mm. crankpin diameter. This is due to the fact that on the one hand large crankshafts are made of a material different from that used for small crankshafts, and that on the other hand the large shafts are made simpler in shape because of reasons of manufacture. Finally, the influence of size must also be considered, which makes it impossible to apply test results from small models to the full sized prototype. Thus, for instance, it was found in a preliminary test that the torsional fatigue strength of a scale model with 40 mm. crankpin diameter was twice as large as that of the full-size crankshaft having a crankpin of 245 mm. diameter, the same material being employed in both cases.

It was therefore considered an urgent necessity to undertake full size tests ; and it was adjudged particularly important to ascertain the cyclic torsional fatigue strength, paying special attention to the influence of flaws of the kind revealed by the magna-flux method. Since supplementary investigations of the bending fatigue strength are still in progress, only a few typical data concerning this problem can be included here.

Before relating the results of fatigue tests, an account may be given here of the most important points to be considered in determining operational stress amplitudes. The crankpin diameter is of foremost importance in deciding upon crankshaft dimensions. On the one hand, it is dependent upon the specific bearing pressure of the big end bearing, which in present day practice is based on a computed peak value of 120–200 kg. per sq. cm. (This computed peak value is arrived at by dividing the product of piston area and maximum gas pressure by the projected area of the crankpin.) On the other hand, the crankpin must be so dimensioned that no torsional natural frequency falls within the main range of operational speed.

On this basis, the rather low nominal torsional stresses of $\tau_m = 150$ to 350 kg. per sq. cm. are arrived at in present day average diesel practice based upon the use of steel St.C. 35.61. The nominal torsional stresses are computed by dividing the torque corresponding to maximum load by the polar section modulus of the crankpin.

From the aspects of crankshaft fatigue strength and danger of fracture, the mean torque and its co-ordinated nominal stress τ_m are only of secondary importance. Of prime importance, however, is the magnitude of the oscillating torques which are superimposed on the mean torque. Magnitude and type of these oscillating torques at the present time can only be estimated either from the theoretical torque diagram or from actual records taken by means of a torsiograph. In this, particular attention must be paid to the nodes of natural frequencies.

With regard to the magnitude of cyclic torsional stresses observed in the operation of crankshafts of the size under consideration, it can be said that they range in amplitude from $\tau_m = \pm 100$ to ± 300 kg. per sq. cm., but these figures are still subject to some revision. But it must be pointed out that no fatigue cracks in large diesel crankshafts were experienced ever since torsional vibration dampers had been fitted. This justifies the conclusion that the amplitudes of torsional cyclic stresses under operating conditions are below the stress values employed in fatigue strength tests conducted on crankshafts subsequent to their manufacture.

Still more important is the computation of the bending stresses to be expected in operation. The crankshaft may be considered as a beam resting on a number of supports subjected to elastic deformations which are caused by distortions of the crankcase. In the case of marine diesels, for instance, such crankcase deformations may be due to distortion of the hull. Such dislocations of the bearings may set up additional bending stresses in the crankshaft. The supporting action of the bearings upon the journals also lacks precise definition, the supporting action of the oil film being an additional factor of uncertainty in this connection. Considerable bending stresses may also be caused by the dynamic unbalance of rotating masses. The magnitude of bending stresses occurring in actual operation cannot therefore even be estimated by computation, but must be ascertained by dynamic strain measurement during actual operation of the engine. Measurements of this nature are exceedingly difficult to make. A first attempt in this direction has, however, been made and will be reported at a future date.

The investigation referred to hereunder was concerned with the classification of the following problems :

(*a*) What is the torsional fatigue strength of a full-size crankshaft and at what point does failure originate ?

(*b*) Is the torsional fatigue strength thus ascertained influenced to any considerable extent by flaws such as are found by the magna-flux test ? Can a diminution in torsional fatigue strength be avoided if fissures at dangerous spots are cut away by a milling cutter, the edges of the cuts being well rounded off ?

(*c*) What is the bending fatigue strength of the crank and in what manner does failure take place ?

TORSIONAL TESTS CONDUCTED ON SECTIONS OF CRANKSHAFTS WITH 245 mm. CRANKPIN DIA. MADE OF STEEL St C 35.61.

The tests were conducted with the use of the device shown in Figs. 1 and 2. Referring to Fig. 1, the oscillating system employed to generate the cyclic

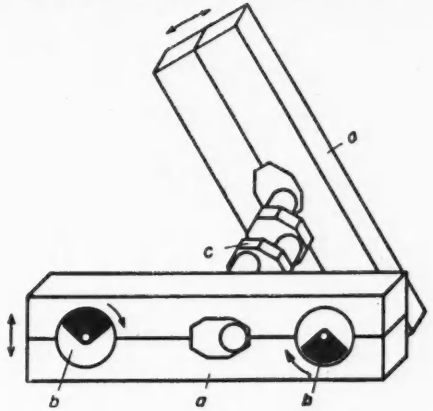


Fig. 1. **Arrangement for testing torsional fatigue strength of large cranks.**
(a) Oscillating mass ; (b) Rotating unbalanced masses ; (c) Crank.

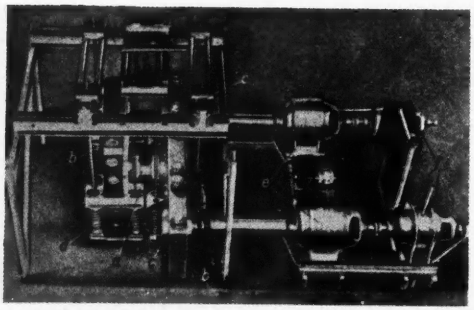


Fig. 2. **Device for determining torsional fatigue strength of large crank shafts.**
(a) Crankshaft section; (b) Oscillating masses; (c) Rotating unbalanced masses setting up oscillations; (d) Suspension springs; (e) Diesel current motors; (f) Three phase synchronous generators.

torsional stresses is seen to consist of two oscillating masses in the shape of 2 pairs of iron bars, one pair each being clamped over webs specially increased in thickness (Fig. 4), in order to permit a firm grip of the clamps and to withstand the considerable surface pressure required to produce a firm seat. This clamped joint was found not to work loose even after more than 100 million oscillations. The oscillating system is operated at a frequency somewhat below its natural frequency. The oscillations are generated by the two revolving unbalanced masses *b* carried in one of the clamps, as shown in Fig. 1. The force diagram given in Fig. 3 illustrates the way in which this device is made to produce a single couple at all times. Both masses are made to revolve in equidirectional synchronism with their respective centres of gravity always enclosing an angle of 180 degrees. By maintaining this phase difference of 180 deg., the horizontal components of the centrifugal force are always kept in equilibrium, while the vertical components are always equal but opposed in direction, thus forming a couple.

The two revolving masses are driven over shafts

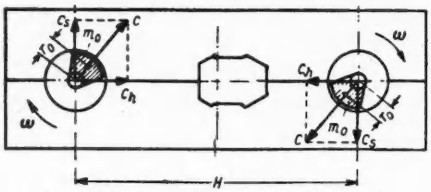


Fig. 3. **Force diagram of the rotating unbalanced masses.** The horizontal component forces C_h are in equilibrium while the vertical component forces set up the couple $M_d = C_s$. $H = m_o r_o . \omega^2 . H . \cos \omega (t)$.

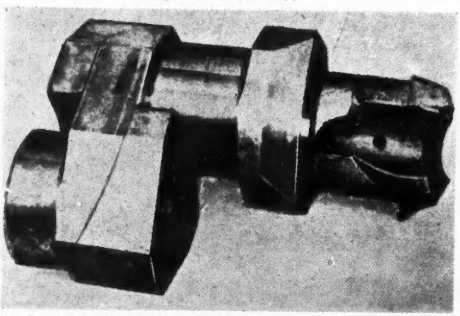


Fig. 4. **Method employed in increasing thickness of web in order to accommodate oscillating clamp.**

with universal joints, as shown in Fig. 2. The two driving electric motors are electrically synchronised, thereby also maintaining the 180 deg. angularity of the revolving unbalanced masses. The diagram of the circuit employed is shown in Fig. 5, where each of the two driving motors of the d.c. shunt type is seen to be coupled with a two-pole three-phase synchronous generator, the slipring brushes of the latter being directly connected with each other. In this way the two generators keep each other in step.

Maintenance of constant torsional stress is a problem particularly difficult to solve, as will be seen by referring to Fig. 6. Here it is shown that a variation in speed of 1 per cent causes a variation in stress of approx. 10 per cent. Therefore if a test accuracy of ± 2 per cent is to be obtained, the speed must not vary by more than ± 0.2 per cent. This was achieved by actuating the automatic controller over a resonance circuit, a speed variation of 1 per cent causing a variation of the control impulse current of 20 per cent. In this way the accuracy of the automatic controller is increased from ± 1 per cent to ± 0.1 per cent, which corresponds to torsional stress variations of no more than ± 1–2 per cent. The torque applied to the crankshaft is measured by an induction type dynamic torsiometer according to E. Lehr, which is attached to the crankshaft. The entire testing device is suspended by a system of helical springs, so that no external restraining or other forces can act upon the system.

The physical data and the composition of the steel of the tested crankshafts are as follows :—

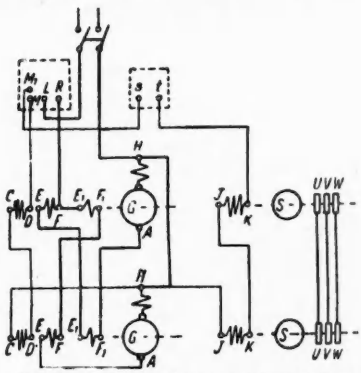


Fig. 5. **Circuit of the synchronous drive of the torsional oscillator employed for the testing of large crankshafts.** G = Diesel current shunt type motors. S = Two-pole three phase synchronous machines coupled to the diesel current motors.

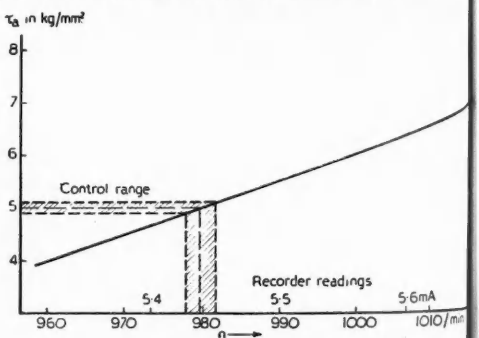


Fig. 6. **Resonance curve extending over the working range.** The shaded areas indicate the central range corresponding to a variation of τ within $\pm$ 5 kg. per sq. mm.

Yield point : 30 kg. per sq. mm.
U.T.S. : 55–65 kg. per sq. mm.
Elongation : 22 per cent.
Impact strength : 6 mkg.
C =0.35–0.4%
Mn =0.8%
Si =0.35%
S =0.04%
P =0.04%
S+P =0.07%

Crank (1) was found to fracture after 2.3 million load cycles under a load of ±7 kg. per sq. mm., the fracture being illustrated in Fig. 7. The fracture is seen to have started from the flaw *A* which had been milled out rather roughly, while the fatigue fracture proper extends at an angle of 45 deg. with the axis of the crankpin. Another crank greatly overstressed by loading at ±9 kg. per sq. mm. failed after approx. 1 million load cycles, the fracture starting from the fillet and also extending at an angle of 45 deg. A third crank was tested at ±4.6 kg. per sq. mm. A first fissure appeared after 7 million cycles, but the final fatigue failure occurred only after 75 million cycles. By charting these three results, a S/N curve was obtained at first approximation, as shown in Fig. 8. This shows that the limiting safe range of stress approximates to ±4.2 kg. per sq. mm. The fourth crankshaft was therefore tested at this stress value and successfully withstood 80 million cycles without showing fissures. Nor did the flaws previously ascertained by the magna-flux method undergo any changes. The stress was then increased to ±5 kg. per sq. mm., a fissure appearing after approx. 25 million cycles. This was found to have its origin in a flaw in the cylindrical part of the crankpin. Failure was ultimately brought about by increasing the stress to ±5.6 kg. per sq. mm., the actual fracture being shown in Fig. 9.

Fig. 7. **Fatigue failure.**
A = Incipient cracks D = fatigue fracture.

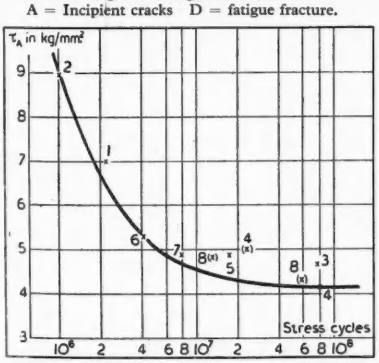


Fig. 8. **S/N curve of crank with 245 mm. crankpin diameter.**

Another test was conducted with a view to ascertaining whether a higher fatigue strength obtains if the piece contains no detectable flaws at all. The fracture obtained in this test and shown in Fig. 10 took place as early as after 4.2 million load cycles, the stress being ±5.5 kg. per sq. mm. As will be seen in the illustration, failure took its inception in the fillet. This goes to

Fig. 9. **Crankshaft fatigue fracture.**
A = incipient fissure. D = fatigue fracture.

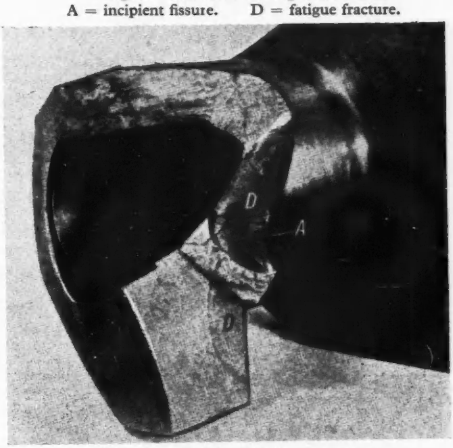


Fig. 10. **Crankshaft fatigue fracture.**
A = incipient fissure. D = fatigue fracture.

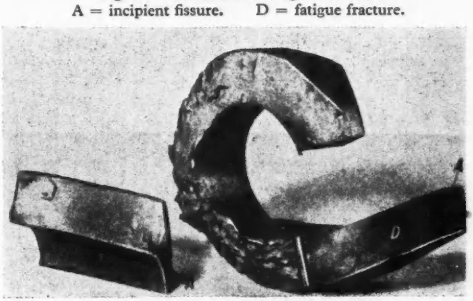


Fig. 11. **Crankshaft fatigue fracture.**
A = incipient fissure. D = fatigue fracture.

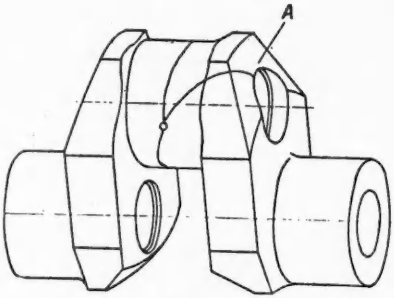


Fig. 12. **Diagram of fracture shown in Fig. 11.**

show that cranks without detectable flaws do not possess superior fatigue strength as compared to pieces containing flaws.

Still another test was concerned with a crank, the pin of which was permeated by numerous flaws distributed not only throughout the cylindrical part, but also throughout both fillets. The test run was made with a loading of ± 4.8 kg. per sq. mm., failure occurring after 8.3 million cycles. As will be seen from Fig. 11, the fracture was quite uninfluenced by any surface flaws. However, in contradistinction to all other tests conducted so far, failure took its inception at the outer edge of the bore of the pin, this edge being particularly endangered in this respect because of the recess provided for the plug closing the hollow bore. Owing to the different point of incipient failure, the entire fracture took a course distinctly different from all others experienced.

Referring to Fig. 12, it will be noted that the fracture passes through the oil hole although this was not centrally placed, as was also the case with the other cranks tested. But it must be assumed that this path of fracture was rather accidental and not induced by stress concentration at the oil hole. This test result must be taken as another proof of the fact, that notches forming part of the design are of greater influence upon fatigue strength than notch effects originating from flaws.

A test was also made to determine the extent to which a reduction in the fillet radius from 15 to 5 mm. reduces the fatigue strength of the crank. Two runs, each covering 30 million cycles with loading of ± 3.2 and ± 3.7 kg. per sq. mm. respectively, were made without incident. After another 17.3 million cycles

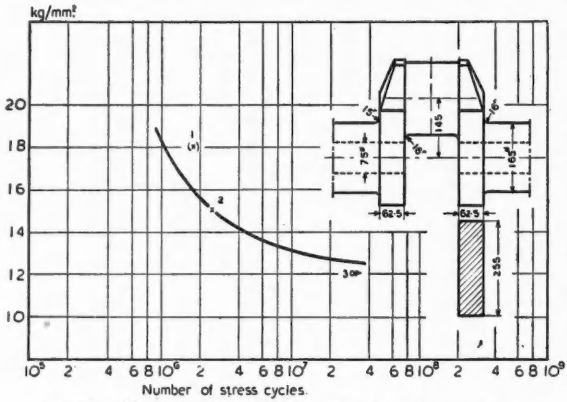


Fig. 13. **S/N curve of crankshaft model with 165 mm. dia. crankpin.**

Fig. 14. **Crankshaft fatigue failure.**
A = incipient crack
D = fatigue fracture.

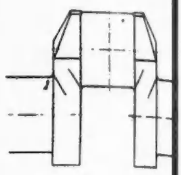


Fig. 15. **Location of fracture shown in Fig. 14.**

with a loading of ± 4.2 kg. per sq. mm., a considerable number of fairly small fissures occurred in the fillet, whereupon the test was broken off. Final fracture was then effected in a power press. From this investigation it can be concluded that the reduction in the fillet radius leads to a reduction in torsional fatigue strength by some 10 per cent.

In addition to these tests the fatigue strength of cranks with a pin diameter of 165 mm. was also determined by testing them in a large rotating bending stress fatigue machine capable of exerting bending moments of up to 20,000 mkg. Up to the present, three cranks were brought to failure by means of this machine. In the case of these cranks the hollow bore of the crankpin was omitted. The test results are charted in Fig. 13, where the S/N curve is also entered. This curve shows that the nominal limiting safe range of bending stress lies at 12–13 kg. per sq. mm., based on the section of the web. This nominal stress was arrived at by dividing the test bending moment by the section modulus of the web. But if the nominal stress is based upon the cross sectional area of the crankpin, a limiting safe range of stress of 4 kg. per sq. mm. is obtained. (It may be mentioned that R. Schmidt,* in testing aero-engine crankshafts of alloy steel of 115–139 kg. per sq. mm. U.T.S., obtained a limiting safe range of stress of ± 22.8 kg. per sq. mm., this stress being based upon the cross sectional area of the web.)

A photograph of the fracture is reproduced in Fig. 14, the type of the fracture being diagrammatically shown in Fig. 15. Contrary to what would be expected, the lines of fracture do not extend from outer to inner fillet, but enter from the fillets into the webs at an angle of 30 deg., and therefore remain discontinuous.

Comparative tests conducted on round bars of the same material (St. C. 35.61) yielded torsional fatigue strength values of 26.0, 16.0, and 18–19 kg. per sq. mm., the respective diameters being 7.5, 30, and 165 mm. Compared to these values, the fatigue strength of the finished crankshaft must therefore be considered rather unsatisfactory; and it should be possible to find ways and means to achieve considerable improvements in this respect.

* R. SCHMIDT, *Luftwissen*, Vol. 9, 1942, p. 263.

THE AACHEN HOT BLAST CUPOLA.

By E. PIWOWARSKY. (From *Die Giesserei*, Vol. 30, No. 20/22, October, 1943, pp. 221-225).

ON several occasions the author has voiced the opinion that in order to achieve a high pouring temperature, a maximum amount of blast should be used. Experimental investigations have, however, shown that there exists the danger of completely chilling the coke if blast rates in excess of 250 cu. m. per sq. m. per minute are employed. It is also known from practical experience that with blast rates exceeding 160 to 180 cu. m. per sq. m. per minute, the losses by depletion show a considerable increase, which ultimately seriously impairs the uniformity of the tapped metal. Slagging of the tuyeres also increases to such an extent that a uniform blast can only be maintained with great difficulty. Furthermore, the higher the ignition point of the coke, the larger will be the cooling and oxidising zone extending immediately above the air belt; the advantages derived from the use of a hard coke difficult to ignite are therefore partly cancelled.

In contrast to this, the use of a hot blast offers considerable advantages, particularly as regards uniformity of operation, with the slagging of the tuyeres also decreasing. The theoretical temperature of combustion of the coke increases according to the expression

$T_{max} = \frac{Q+W}{R \cdot c}$ where Q is the calorific value of the fuel expressed in cal. per kg., W is the heat content of the hot blast in cal. per kg. of coke, R is the quantity of products of combustion in normal cu. m. per kg. of coke, and c is the specific heat of the combustion products.

According to R. Dawidowski* the expression $T_{max} = \frac{W}{G \cdot C}$ can be used, as this is accurate within 2 per cent; and it can be seen that high combustion temperatures are possible even with coke of comparatively low hardness, provided the blast temperature is chosen high enough. In cupolas of the Griffin, Schurmann, and other well-known types, the sensible and latent heat of the products of combustion is utilised for preheating the blast to temperatures ranging from 150 deg. C. to 300 deg. C. max. Where only the sensible heat of the gases is utilised, blast temperatures do not exceed 100 deg. C. to 150 deg. C. (Moore Cupola, American Cast Iron Pipe Co.). Heretofore temperatures in excess of about 300 deg. C. were not, however, aimed at. As a matter of fact, the well-known U.S.A. Patent No. 1,367,479 of 1921 expressly limits heating of the blast to a maximum temperature of 500 deg. F. (290 deg. C.). This attitude was probably due to the belief that higher blast temperatures would cause an undue increase in CO formation, thereby jeopardising the melting operation. This opinion was also strongly voiced by B. Osann,† and unfortunately it is still shared today by many foundry experts.**

In the hot blast cupola proposed by the author, high combustion temperature is obtained with the employment of a hot blast with the air temperature ranging preferably from 450 deg. C. to 600 deg. C. Only by employing a hot blast of this temperature range and by the reduction of the amount of coke per charge to 8 per cent, is it possible to exploit the inherent advantages of hot blast operation. In this case the blast is to be heated by a separate heating device.

It is true enough that with the employment of a very hot blast in conjunction with an unchanged percentage of coke in the charge, the higher operating temperature

* R. DAWIDOWSKI: The advantages of using hot blast in cupola operation. *Intern. Giess. Kongr. Warschau*, 1938.

† B. OSANN, *Stahl & Eisen*, 1908, Vol. 28, p. 1497; see also *Giesserei Praxis*, 1936, Vol. 59, p. 436.

** *Giesserei*, 1941, Vol. 28, p. 333.

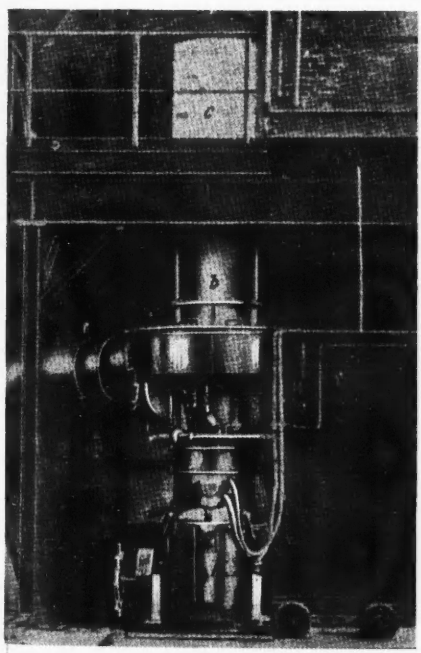


Fig. 1. **View of Aachen hot blast cupola.**

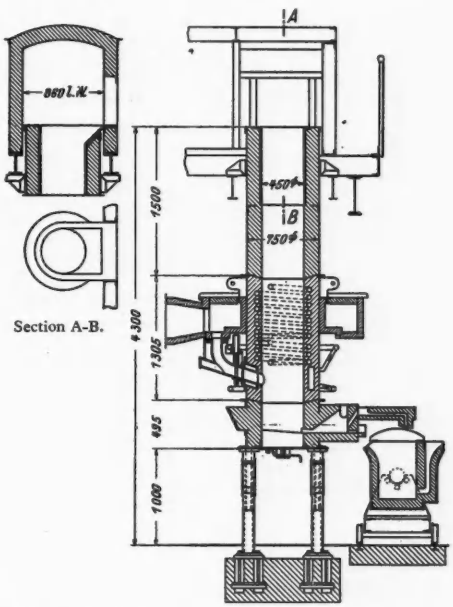


Fig. 2. **Sectional elevation of Aachen hot blast cupola.**

tends to increase the CO content of the products of combustion at the stack. In the Aachen hot blast cupola, dissociation of the CO_2 under the influence of the hot blast is almost completely counteracted by the opposing influence of the diminution in the amount of coke in the charge. According to the temperature of the hot blast, this diminution in the coke quantity may amount to one to three per cent, as compared to that required with cold blast. Actually, with the use of a good grade of coke the diminution is usually found to amount to some 7–8 per cent. In the case of the experimental cupola in which the large-sized coke in the charge amounted to 10 per cent, even with 500 deg. C. hot blast temperature, 18 per cent CO_2 and 5 per cent CO could still be achieved.

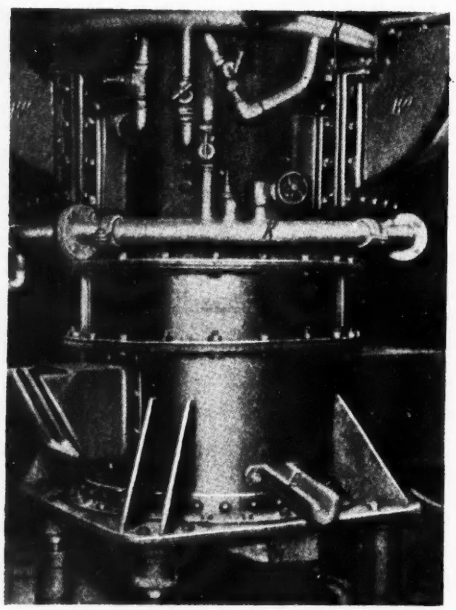
The first experimental runs showed that a very hot metal of excellent pouring properties and of low sulphur content is obtainable. The melting losses practically equal, or only slightly exceed, those registered with cold blast operation. Also, the output can be doubled or even further increased, as compared to cold blast operation. Furthermore, the uniformity of the entire melting process is improved, since no slagging of the tuyeres occurs if the hot blast temperature is kept above 400 deg. C. Generally, the melting zone lies somewhat lower in hot blast operation, so that a greater furnace depth is required where considerable amounts of recarburisation are to be obtained. In itself, however, a higher degree of recarburisation can be effected with hot blast operation than with a cold blast. The thermal efficiency of the hot blast cupola ranges from 55 to 65 per cent.

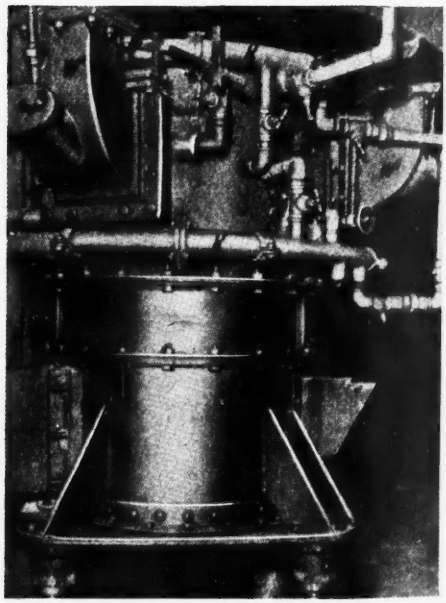
When operating with a reduced coke percentage in the charge, full advantage of the achievement of higher metal temperature can only be taken by resorting to special modes of operation. For the actual achievement of high metal temperature, the coke percentage should therefore be increased to some 12 per cent of the charge. With a hot blast of 500 deg. C., the CO content of the waste gases will then be so high that they can be burned with the hot blast, serving for instance, for heating the mixer.

Owing to the high temperatures produced in the melting zone by the hot blast, efficacious protection of the lining against excessive wear is imperative. For this purpose, the first hot blast cupola installed at the Aachen Foundry Institute was equipped with six cooling coils placed in the refractory at a point above the tuyeres, each cooling coil having its own cooling water supply. Cooling water consumption proved not to be excessive, and the refractory stood up well. Considerable wear of the lining, however, occurred at the tuyere level, this being due to the fact that with hot blast operation, the zone of highest temperature lies at a lower level as compared with cold blast operation. Thus with a hot blast of 600 deg. C. the zone of maximum temperature lies only slightly above the tuyere plane, as the temperature of the blast almost equals the ignition point of the coke.

The method followed in effecting efficacious cooling of this zone is shown in Fig. 2. Three water-cooled copper-tuyeres are installed, with cooling boxes placed between them. Practically the entire furnace circumference is then water cooled. On their face, these boxes, as well as the cooling coils, are covered with a 50 mm. thick layer of plastic refractory cement. So far this cooling system has proved satisfactory, although some minor changes may have to be made at a later date, such as making the water cooled tuyeres project some 10–20 cm. into the furnace.

The Iron and Steel Works (formerly Georg Fischer) at Schaffhausen, Switzerland, were the first to install in 1939, a hot blast cupola plant, the operation of which was begun in Feby., 1941. An outline of this installation equipped with a furnace of 650 mm. diameter, is given in Fig. 5. This cupola produces malleable iron, with the coke percentage of the charge amounting to 6 per cent, as compared to 13 per cent with cold blast operation. The hot blast is heated to 450–600 deg. C. in a Sicromal heater. The sulphur content of the tapped metal was found to decrease slowly with hot blast operation, while formerly the high coke percentage and the re-melting of abt. 50 per cent sprue had led to a slow increase in sulphur. As the subjoined heat





Figs. 3 and 4. **Air belt and base of cupola, W1=hot blast supply ; K=cooling coils.**

Heat Balance of Cupola of 650 mm. Diameter Producing Malleable Iron.

	With Cold Blast	With Hot Blast of 300 deg. C.	With Hot Blast of 600 deg. C.
Heat content of coke used, %	97.0	88.7	79.3
Heat for combustion of impurities, %	3.0	3.9	4.9
Heat content of hot blast, %	—	7.4	15.8
Total ..	100.0	100.0	100.0
Heat content of the molten iron %	31.8	51.8	60.0
Heat content of the molten slag, %	3.34	6.4	5.65
CO_2 decomposition of the limestone, %	1.6	2.07	—
Evaporation of moisture content of coke, %	0.02	0.05	0.04
Decomposition of moisture in blast, %	1.9	2.02	3.06
Sensible heat of products of combustion, %	9.2	5.3	4.95
Heat content of products of combustion (incomplete combustion), %	43.0	26.5	20.0
Cooling water loss, ..	0.94	0.8	1.9
Total ..	100.0	100.0	100.0
Thermal efficiency of cupola, %	31.8	51.8	60.0
Thermal efficiency of blast stove, %	—	60.0	64.2
Thermal efficiency of entire plant, %	31.8	48.7	53.8
Temperature of products of combustion, deg. C. ..	350	200	200
Total coke consumption in kg. per ton of molten metal ..	163.0	106.0	79.0
Anthracite for hot blast stove, kg.	—	27.0	24.8
Coke and anthracite, in kg. per ton of molten metal ..	—	133.0	103.8
Saving in coke, compared to cold blast, %	—	35.0	51.5

balances show, hot blast operation of 450–600 deg. C. yields effective savings in coke of 18–38 per cent. Part of this saving must be credited to the high efficiency of the Sicromal heat exchanger, which ranges from 75–85 per cent at full load. In the plant considered, the heat exchanger is fired with producer gas. With hot blast operation, melting losses were found to be somewhat higher than with cold blast operation; but the tapped metal proved to possess excellent teeming and casting properties, although its temperature was no more than 30 to 50 deg. C. higher than with cold blast operation.

The systematic tests referred to above furnish proof of the considerable advantages—both with regard to quality of the metal and operation of the plant—accruing from the use of hot blast with temperatures ranging from 300 to 600 deg. C. Contrary to previous conceptions held by the majority of experts, cupola operation with high blast temperatures is in principle, therefore, both feasible and advantageous, thus vindicating the contentions of the author

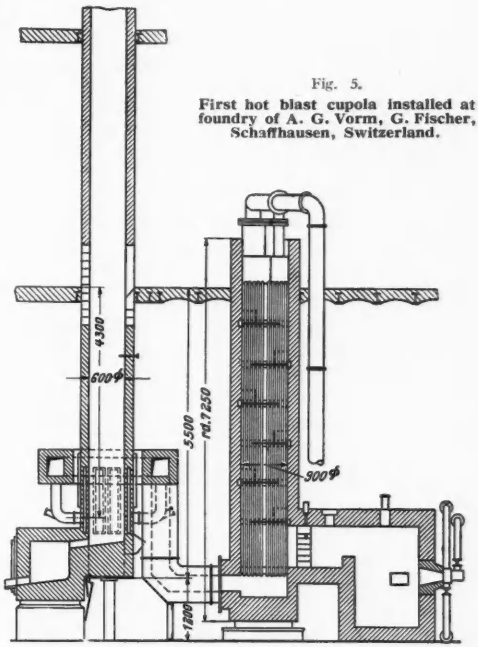


Fig. 5.
First hot blast cupola installed at foundry of A. G. Vorm, G. Fischer, Schaffhausen, Switzerland.

Encouraged by the good results obtained with the first small hot blast cupola installation, the Iron and Steel Works (formerly Georg Fischer) have converted four cupolas of about 900 mm. furnace diameter to exclusive hot blast operation. Since September, 1941, these cupolas have produced many thousand tons of malleable iron. Similar plants are now under construction in large German foundries.

MODERN HARDNESS TESTING MACHINES.

By Dipl. Ing. K. MEYER. (From *Fertigungstechnik*, Vol. 1944, No. 2, February, 1944, pp. 47–51.)

OPTICAL HARDNESS TESTERS WITH INTEGRAL MICROSCOPE.

A WELL-KNOWN hardness testing machine with integral microscope is the *Brivisor* 3000 machine shown in Figs. 1 and 2, which is suitable exclusively for Brinell and Vickers tests. Its testing range comprises the standardised loadings between 60 and 3000 kg., and loadings of 60, 80, 100 and 120 kg. for Vickers tests. The individual loading steps are obtained by appropriate combination of deadweights. The loading device consists of two fulcrum-supported levers with the main lever projecting from the housing, while the loading lever in the housing carries the deadweight as shown in Fig. 2. Lifting of the levers is effected in well-known manner by the action of the adjustable dashpot (b), which affords an easy control of the rate of loading. The projecting end of the main lever carries a slide for reception of the penetrator head. This slide can be moved by hand into two extreme positions, for either permitting the use of the microscope or for bringing the penetrator head into position for making the indentation. In the latter position, the direction of loading coincides with the optical axis of the microscope. The latter is a special instrument of 25 × magnification provided with an obliquely arranged eye-piece and equipped for dark-field illumination, that is, the impression is seen as a bright area in a dark field. The eye-piece contains a vernier device formed by two graduated glass plates, each moved by a thumb screw. This permits an accuracy of reading of 0.001 mm. In order to be able to carry out rapid examinations, tolerance marks are also provided, which can be set at the respective tolerance limits required.

The *Brivisor* 250 is a spring loaded machine suitable for loadings from 10 kg. to 250 kg., while the *Brivisor* 62.5 permits spring loadings from 3.91 kg. to 62.5 kg. (Figs. 3 and 4). Because of the smaller impressions obtained with these machines, they are equipped with microscopes affording a 50 × magnification. In all other respects these machines are similar to the *Brivisor* 3000. They are particularly suited to

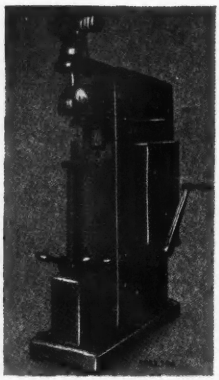


Fig. 1. **Hardness tester BRIVISOR 3000 with fitted microscope.**

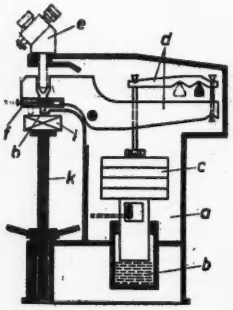


Fig. 2. **Hardness tester BRIVISOR 3000 with fitted microscope.**
(a) Housing ; (b) Dash pot ; (c) Deadweights ; (d) Loading lever ; (e) Microscope ; (f) Slide ; (h) Penetrator ; (i) Test piece ; (k) Table spindle with hand wheel.

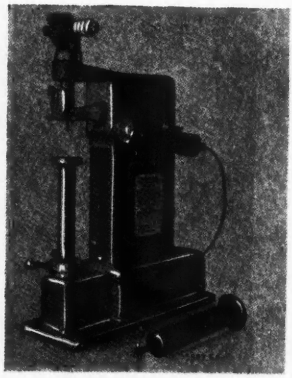


Fig. 3.
Hardness tester BRIVISOR 62.5.

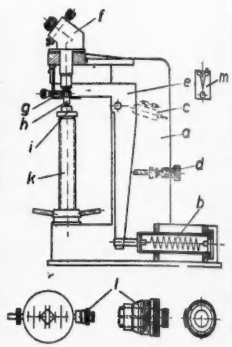


Fig. 4. **Hardness tester BRIVISOR 62.5.**
(a) Housing ; (b) Loading spring ; (c) Illumination ; (d) Stop for load setting ; (e) Loading lever ; (f) Microscope ; (g) Slide ; (h) Penetrator ; (i) Test piece ; (k) Table spindle with hand wheel ; (l) Measuring device ; (m) Loading adjustment.

carrying out Vickers tests, the *Brivisor* 250 providing all standardised Vickers loadings between 10 kg. and 120 kg., while the *Brivisor* 62.5 is equipped for all Vickers standard loadings between 5 kg. and 60 kg. The loading device of these machines consists of an angular lever, one of its ends projecting from the housing and carrying the slide for the penetrator head, while the other is connected to the loading spring housed in a cylindrical tube. Both machine types are furnished with sets of loading springs corresponding to the standardised Vickers and Brinell loadings falling within their working range. Their operation is similar to that of the *Brivisor* 3000 type.

The rapid ball indentation press *Km* 3 is a single purpose machine which can be used for standard Brinell tests only ; its loading range extends from 3000 kg. to 250 kg., and in special cases to 187.5 kg. The loading is effected by deadweights so subdivided that all standard loadings within the working range can be obtained. The loading ram is furnished at its top with a seat for the knife edge of the loading beam, while the lower end of the ram is guided by a ball bearing.

The measuring device of the microscope consists of a movable dial placed in the eye-piece with gradations of 0.05 mm. The dial, which is moved by a thumb screw, can be rotated 90 deg. so that the diameter of the impression can be measured in various directions. This rapid ball indentation machine is built in two models, namely type *KGm* 3 (Fig. 5) for manual operation and type *KEm* 3 with electrical drive (Figs. 6 and 7).

In the manually operated type the lifting and the release of the deadweights is effected by lever and crank, the speed of descent of the deadweights being adjustable by means of a dashpot control valve. Unloading after completion of the indentation is carried out by manual depression of the operating lever. The deadweights of the electrically operated machine are lifted and lowered by the action of an electric motor driving a crank through a worm gear and a two-speed reduction gear. The latter permits the choice of 10 sec. or 30 sec. loading time. The hardness testing of the individual leaf of a leaf spring is shown in Fig. 6. In this test a special support with spherical surface is placed on the table spindle, the leaf being held in place by pressing it against the lower edge of the casing containing the penetrator head.

The rapid ball indentation machine *KCeSm* 3 (Figs. 8 and 9) is used where Brinell tests are to be carried out in great numbers. The loading range of this cantilever type machine extends from 500 kg. to 3,000 kg., hydraulic loading being employed. The standardised loadings of 3,000, 1,000, 750, and 500 kg. are obtained by the setting of a patented pressure-gauge type pressure controller which keeps the loading constant during the test. The hydraulic loading is carried out by means of a piston which moves in the head of the machine ; the head serves as hydraulic cylinder, an electrically

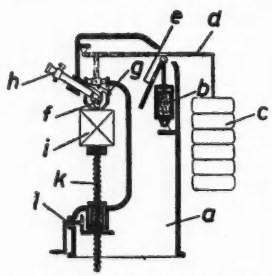


Fig. 5. **Rapid hardness tester KGm 3 for manual operation.**
(a) Housing ; (b) Dash pot ; (c) Deadweights ; (d) Loading lever ; (e) Loading release ; (f) Ram with penetrator ; (g) Lever for tilting (f) ; (h) Microscope ; (i) Test piece ; (k) Table spindle ; (l) Table spindle drive.

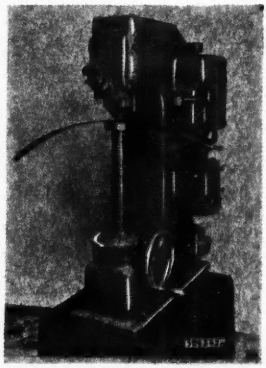


Fig. 6. **View of electrically operated rapid hardness tester KEm 3.**

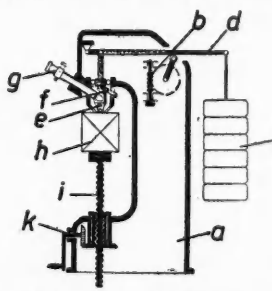


Fig. 7. **Sectional view of electrically driven rapid hardness tester KEm 3.**
(a) Housing ; (b) Electric loading release ; (c) Deadweights ; (d) Loading lever ; (e) Ram with penetrator ; (f) Lever for tilting (e) ; (g) Microscope ; (h) Test piece ; (i) Table spindle ; (k) Table spindle drive.

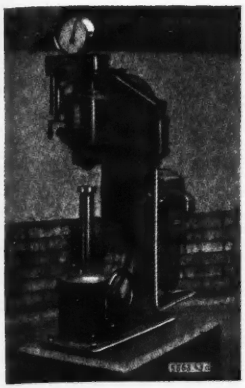


Fig. 8.
Rapid hardness tester KCm 3.

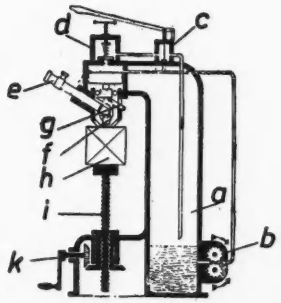


Fig. 9. **Rapid hardness tester KCeSm 3.**
(a) Housing (serving as pressure fluid reservoir); (b) Gear pump; (c) Stop valve; (d) Pressure limiting device; (e) Microscope; (f) Ram with penetrator; (g) Lever for tilting (f); (h) Test piece; (i) Table spindle; (k) Table spindle drive.

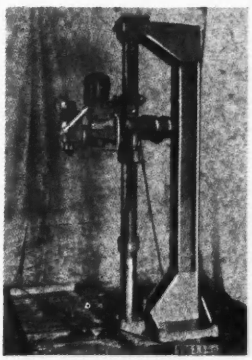


Fig. 10.
Brinell hardness tester KCeSm 3.

driven pump supplying the pressure fluid. Accelerated lifting of the loading piston after completion of the loading is achieved by withdrawing the pressure fluid from the loading cylinder under suction.

This rapid ball indentation machine is also built in two special types of which that shown in Figs. 10 and 11 is intended as single purpose machine for Brinell hardness tests. In this design, which is of the radial type, both loading head and pressure oil pump are arranged on the radial arm, which can be raised or lowered by means of a motor driven spindle as shown in Fig. 11. Upper and lower limit switches are provided. The loading range of this type extends from 500 to 3000 kg. This machine is especially suited to test bulky pieces, which are brought into position on a heavy bogie or on a movable table.

For loading pressures of 1–250 kg., the hardness tester type *KRV* 0.25 with integral measuring microscope can be used. This machine is suitable for Brinell, Vickers and Rockwell tests. For the Rockwell tests the usual dial-type depth gauge is provided. This tester represents the only German multi-purpose hardness testing equipment for measuring both impressed area and depth of impression, in addition to the *Dia-Testor 2R* of the image screen type. It can be used equally well for single tests and for mass production purposes. The loading is carried out by a number of deadweights covering the entire number of standard loadings of the three testing methods. In contrast with most of the other German multi-purpose testers, the minor load is not obtained by upward movement of the table, but is produced by a special deadweight applied to the loading ram while it is still in its home position.

When using the machine for Brinell and Vickers tests, particularly for small loadings, the depth gauge is preferably put out of action by means of a simple device

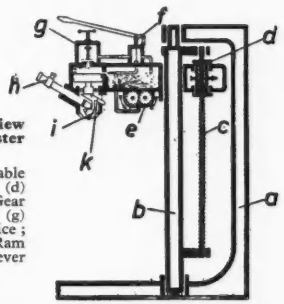


Fig. 11. **Sectional view of Brinell hardness tester KCeSm 3.**
(a) Housing; (b) Rotatable column; (c) Spindle; (d) Spindle drive; (e) Gear pump; (f) Stop valve; (g) Pressure limiting device; (h) Microscope; (i) Ram with penetrator; (k) Lever for tilting (i).

in order to eliminate the influence of the variable pressure of the gauge. These feeler pressures are large enough to increase the loading pressure beyond the tolerance of loading accuracy. Unfortunately, this fact is hardly ever kept in mind in practice. While with a test loading of 1 kg. the officially specified tolerance amounts to plus or minus 10 g., the feeler pressure may be a multiple of this figure. To this must be added the fact that because of the leverage employed, the feeler pressure is multiplied five times.

For the small loading range no hardness testing machines of this type, especially such suitable for mass testing, have developed as yet. In view of the small impressions produced with small loadings, the evaluation of the impressions must be carried out with much greater care than with higher loadings. The length of time required for carrying out these observations as well as the need for specially skilled personnel, preclude, therefore, the application of hardness testing to mass production purposes, at least for the time being, to this loading range.

STRESSES, DEFORMATIONS AND VIBRATIONS OF A CONICAL WING SHELL

By A. PFLÜGER, Focke-Wulf Aircraft Co., Bremen. (From *Luftfahrtforschung*, Vol. 20, No. 1, Jan., 1943, pp. 29-32).

THE engineering theory of bending as applied to beams of full cross-section has proved to be not sufficiently accurate when applied to thin-walled beams such as the load-carrying part of a wing. It becomes especially necessary to pay attention to the change in cross-section due to taper. While with the full cross-section the beam at each station along the beam axis has been replaced by a cylinder of equal cross-section, with thin-walled sections the beam is to be approximated by a cone. The relations derived for the wing-shell can be applied to other conical shell structures such as fuselages or fins.

The discussion of the problem is centred on a cantilever wing (Fig. 1), the load-carrying part of which is a two-spar box beam. The box is the cone, or is to be replaced by the cone. In the following, the box, unless already conical, is replaced by a cone where it is assumed that the wall-thickness will be distributed in such a way that there exists a line passing through the centroids of cross-sections at right angles to it. This

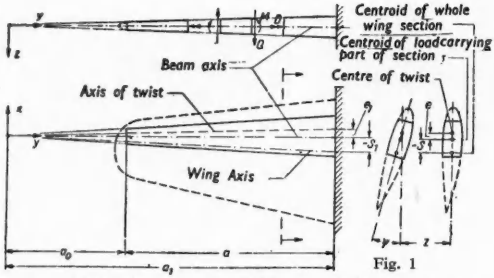


Fig. 1

line we shall refer to as the "beam axis." A second line, defined in the same manner with regard to cross-sections of the whole wing and not of the box only, will be called the "wing-axis." Finally, with the assumptions made for the existence of the beam axis, the centres of twist of cross-sections at right angles to the beam axis will also lie in a straight line through the apex, called the "axis of twist."

The notation used is clear from Fig. 1. The origin of the co-ordinate system is the apex; index o denotes free end of box, index l the restrained end. Ψ is the angle of rotation, $\eta = \frac{y - a_o}{a}$. Further, $E\,I$ is flexural stiffness about the principal axis of the section parallel to x, $G\,I_D$ is the torsional stiffness, M the bending moment, the vector of which is parallel to the x-axis. D is the torque referred to an axis through the centres of twist and parallel to the beam axis, Q the shear force parallel to the z-axis.

The derivation of the state of stress in the wing are omitted here as they can be found in the author's paper in *ZAMM*, Vol. 21 (1942), p. 99. The results are given below.

The direct stresses in the skin of the shell can be found in the usual way by dividing the bending moment M by the section modulus.

However, in finding the shear stresses the standard formula can be used only if instead of the shear force Q the value $Q - \frac{M}{y}$ is used, and instead of the torque D the value $D - \frac{e}{y} M$ is substituted. The co-ordinate e of the centre of twist can be found by usual methods as with a cylindrical shell.

In Fig. 2 the ratio of the reduced shear force and the actual shear force $\frac{Q - \frac{M}{y}}{Q}$ is plotted against η.

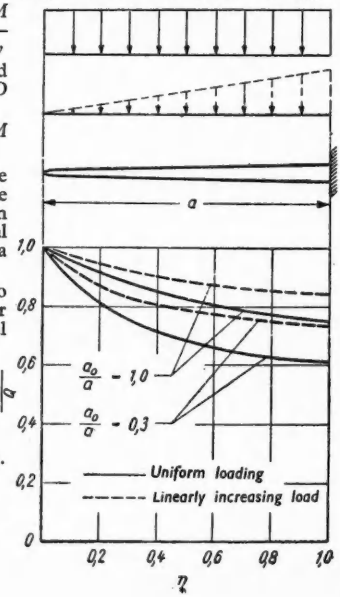


Fig. 2

The parameter is $\frac{a_o}{a}$, which for all practical purposes is between 0.3 and 1.0. From this plotting it appears that the reduction of shear force and its influence on the shear stress cannot be neglected.

To estimate the influence of the reduction of torque by the amount $\frac{e}{y} M$, the following reasoning will suffice: In practical cases the value of $\frac{e}{y}$ is between the limits ± 0.02. As M is much greater than D it follows that with the shear stresses due to torsion the influence of the conical shape of the shell is considerable.

For calculating the vibrations the knowledge of the stiffness of the shell is often desirable. Therefore, we have to deal with the deformation of the shell.

The differential equation of the line of twist is given by

$$\Psi' = \frac{1}{G\,I_D}\left(D - \frac{e}{y} M\right) \quad .. \quad .. \quad (1b)$$

where the stroke refers to derivatives with respect to y. Eq. (1b) shows that there is a rotation of the wing if the bending moment alone acts. There must therefore be a curvature Z'' for torque alone. For the differential equation of the deflection curve,

$$\Psi = \Psi_D + \Psi_M \quad \text{and} \quad Z = Z_D + Z_M.$$

Ψ_D and Z_D be the deformations for $D \neq O$, $M = O$, while Ψ_M and Z_M be the deformations for $M \neq O$, $D = O$.

According to (1b)

$$\Psi' = \frac{D}{G\,I_D} \qquad \Psi'_M = -\frac{e}{y}\,\frac{M}{G\,I_D}$$

For Z_M there is the well-known formula

$$Z''_M = -\frac{M}{E\,I},$$

where $E\,I$ is the flexural stiffness calculated as for the ordinary cylindrical shell.

By using Betti's theorem

$Z''_D = -\frac{D}{M}$, and $\Psi'_M = \frac{e}{y}\,\frac{D}{G\,I_D}$ is obtained.

Then, the differential equation of the deflection curve is

$$Z'' = -\frac{M}{E\,I} + \frac{e}{y}\,\frac{D}{G\,I_D} \quad .. \quad .. \quad (1a)$$

Substituting

$$\overline{E\,I} = \frac{E\,I}{1 - \frac{e^2}{y^2}\,\frac{E\,I}{G\,I_D}}, \quad \overline{G\,I_D} = \frac{G\,I_D}{1 - \frac{e^2}{y^2}\,\frac{E\,I}{G\,I_D}}$$

and with some transformation, (1a) and (1b) yields

$$\left.\begin{aligned} M &= -\overline{E\,I}\left(Z'' - \frac{e}{y}\,\Psi'\right) \\ D &= G\,I_D\,\Psi' - \overline{E\,I}\,\frac{e}{y}\,Z'' \end{aligned}\right\} \quad .. \quad (2a, b)$$

Due to the smallness of $\frac{e}{y}$, $\overline{E\,I}$ and $\overline{G\,I_D}$ can always be replaced by $E\,I$ and $G\,I_D$.

The vibrations of a conical shell are derived in the original of the present abstract, but omitted here as the author's conclusion is that for purposes of vibration the influence of the conical shape can be neglected.

TAPER MEASURING DEVICE

(From *Die Werkzeugmaschine*, Vol. 47, No. 6, March, 1943, pp. 136-138).

In machine shop practice the measurement of cones, prisms, tapered threads, and bodies of similar contour is frequently required. It is well-known that considerable difficulties exist in measuring bodies of this kind, these difficulties being due to the absence of parallel reference lines which would permit the application of instruments or gauges possessing parallel measuring surfaces. If the measurement of a taper is carried out with rollers and bar gauges, the line of actual measurement is displaced, which leads to tedious computations in order to ascertain the actual dimensions. This and other still more involved problems of measurement, which cannot be solved with the use of any other measuring device, can be easily solved by the employment of the completely novel taper measuring device shown in Figs. 1-3.

This device makes it possible to rapidly ascertain the dimensions of tapered bodies by direct high-precision measurement, no computations being required at all. The chief feature of the instrument is the provision of self-adjusting rotatable feeler pads, their axes of rotation being tangential with the piece to be measured. Referring to Fig. 1, the instrument is seen to have a sturdy base, equipped with levelling screws. On top of it rests the hardened and lapped base plate of the instrument proper. This plate carries a vertical pillar with the travelling slide support.

Rapid adjustment of the height of the slide support is effected by a push button controlled electric motor which drives the threaded spindle recessed in the pillar, the spindle carrying the travelling nut of the slide. The final level adjustment is made by means of a micrometer screw arranged at the front of the base. The horizontal measuring slide proper is carried on the slide

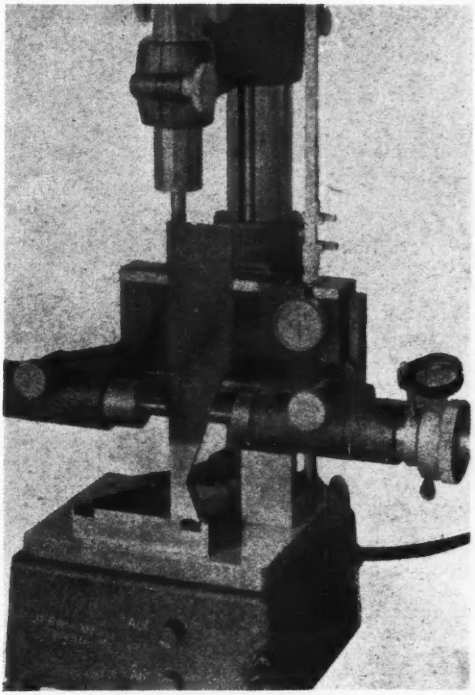


Fig. 2. **Taper measuring instrument. Measuring a slant.**

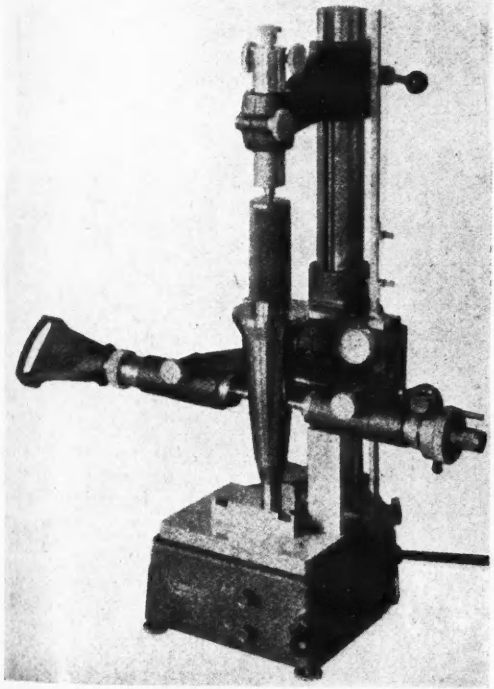


Fig. 1. **Instrument measuring the taper of a conical piece.**

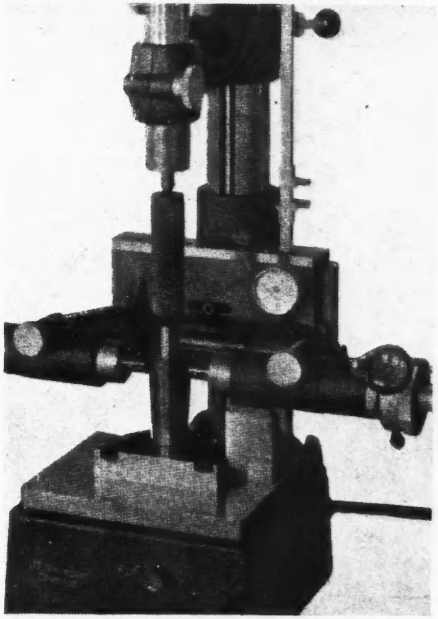


Fig. 3. **Instrument measuring the taper of a conical screw spindle.**

support by means of ball bearings, and has two opposing headstocks which contain the gauging plug holders which can be clamped in position.

The gauging plug on the left-hand side is slidably coupled with a feeler lever, while the right-hand side gauge plug is equipped with a micrometer screw. The micrometer dial is frictionally attached to the micrometer head, permitting it to be rotated into the zero position in which it can be locked by means of a lever conveniently placed for operation.

The two gauge pads consist of half-cylinders of hard metal fitting without play into the troughs of the gauge plugs, the latter being connected with an horizontal millimeter scale. The vertical position of the slide, defined as the elevation of the horizontal axis of the gauge plugs above the aforementioned lapped surface of the base plate, is read off on a vertical millimeter scale; while a feeler gauge with dial is also incorporated in the travelling slide to work in conjunction with slip gauges which may be placed upon the surface of the base plate. A spirit level, a magnifying glass for reading off the micrometer dial, and various other ancillary parts complete the list of equipment.

If, for example, the diameter or width of a given component is to be measured at a certain level, the slide is moved to the approximate elevation by push button control and the final adjustment is then made by means of the aforementioned micrometer screw. The measuring pads are then brought into contact by rotating the micrometer screw of the right-hand side gauge plug, until the gauge pads touch the sides of the body, automatically adjusting themselves to the prevailing slant. The distance between the gauge plugs, that is, the diameter or width of the measured component, is then given directly by the readings of the millimeter scale, the micrometer dial, and the feeler gauge dial. By ascertaining the diameter of the piece at two elevations, the taper is easily determined.

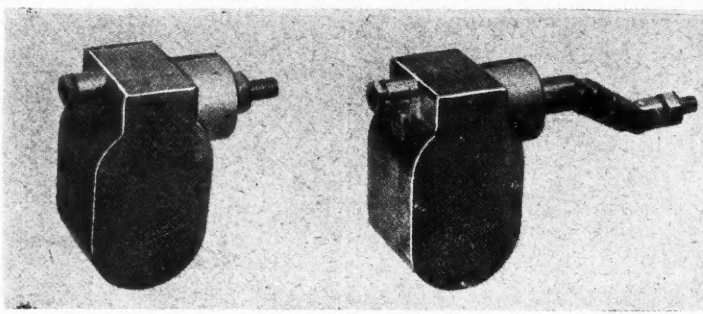
NEW CONTACT ELEMENTS FOR CENTRALOGRAPH INSTALLATIONS.

By S. HJELM. (From *Ericsson Review*, Stockholm, No. 1, 1944, pp. 8–11)

A CENTRALOGRAPH installation is a supervisory system which serves for the continuous supervision from a single observation point of the operation of machine tools, or any other type of machine or mechanical device of a factory or plant. An installation of this kind therefore consists of one or several annunciator installations which receive their actuating impulses from contactors or impulse senders which are mounted on the machines concerned and are operated by these by some mechanical means.

In the latest designs of contact devices developed during the last few years particular stress was laid upon robustness of design in order to ensure satisfactory working under rough conditions of service. Most types are designed, therefore, with cast iron housings.

The rotary contact device shown in Figs. 1a and 1b, which has been employed most frequently, is connected to a revolving shaft of the machine to be supervised. It comprises a die-cast structure of light metal and a gear box with a pressed bakelite cam wheel, which acts upon a group of contact springs. With the use of different gears and cams 32 different ratios between machine speed and impulse frequency can be obtained. The driven shaft of the contact device is carried in ball bearings with the axial thrust being taken by a thrust bearing. The gears operate completely submerged in oil in order to minimise wear. A special oil with very high viscosity is used, and once the apparatus is installed no maintenance is required. The contact device may be directly coupled to the revolving shaft of the machine, as shown in Fig. 1a, or where this is not possible, a universal joint may be interposed as illustrated in Fig. 1b. This contact device is employed where, for the purpose of supervision, it is required to ascertain whether or not a machine is running, what its respective working and idling periods are, or whether a machine operates at the specified speed.



a *b*

Fig. 1. Rotary contact device.

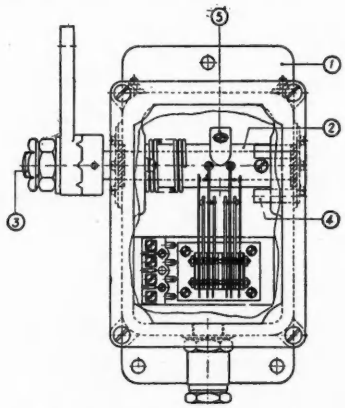


Fig. 2. Lever-operated contact device. Large model.

Where linear motions are to be supervised, lever operated contact devices are employed. The largest type of this kind, mostly used with saw mills, is illustrated in Fig. 2. The device proper comprises a cast iron housing (1) containing a hollow shaft (2) carried in moisture proof bearings. Through the bore of the hollow shaft extends another shaft (3), which is held in place and coupled with shaft (2) by means of a set screw, shaft (3) carrying the operating lever.

It is easily seen that this arrangement makes it possible to remove shaft (3) and to insert it from the other side whenever the lever is required at the right hand side. Also, by means of the coupling shown, the lever can be adjusted to any operating position required. The arc

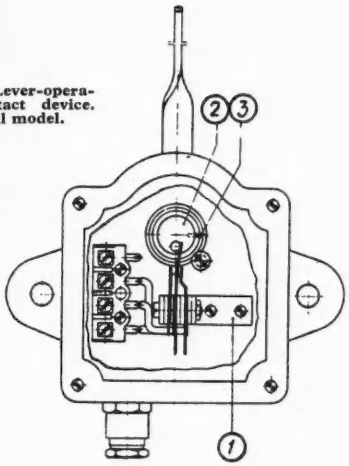


Fig. 3. **Lever-operated contact device. Small model.**

described by the lever can be set by means of two stops (4), and a helical spring for pulling the lever back into its home position can also be provided, as shown. The contacts are operated by means of a bakelite cam (5) affixed to the hollow shaft. With the lever set to an operating angle of 30 deg., a force of 800 g. at the free end of the lever is required to close the contacts.

The somewhat smaller contact device shown in Fig. 3 serves for the supervision of reciprocating motion such as the operation of presses or shears. Here the contacts (1) are operated by a screw eccentrically arranged at the face of shaft (2). A spiral spring (3) is provided to return the shaft into the home position. The operating lever is affixed to the other end of the shaft external to the cast-iron housing. With an operating angle of 45 deg., a force of 100 g. at the free end of the operating lever is required for closing the contacts.

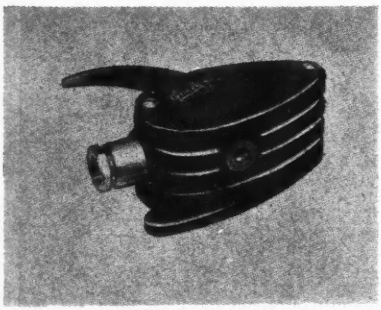


Fig. 4. **Microcontact.**

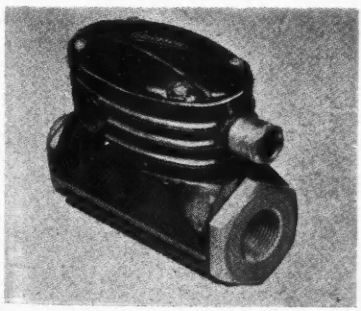


Fig. 5. **Pneumatic contact.**

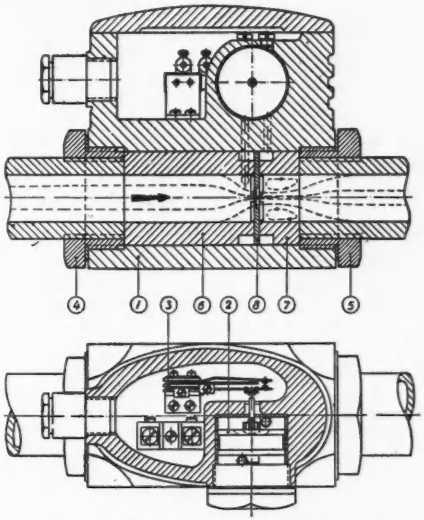


Fig. 6. **Details of pneumatic contact device.**
(1) Cast iron casing ; (2) Stainless steel piston ; (3) Contact spring ; (4) Coupling ; (5) Coupling ; (6) Inlet side ; (7) Outlet side ; (8) Orifice.

The microcontact shown in Fig. 4 is contained in a "streamlined" casing. This contact device is used where small operating forces are available. The position of the operating lever is not adjustable. A force of 8–10 g. at the free end of the lever is sufficient to close the contacts. Another type of microcontact is actuated by a push button with a free travel of 0.4 mm., an operating force of 20 g. being required.

The pneumatic contact illustrated in Fig. 5 is used for the supervision of compressed-air operated tools, such as chisels, grinders, or air spray pistols. These contacts are operated by the pressure drop across an orifice installed in the air supply line of the tool to be supervised. The details of the mechanism are shown in Fig. 6. A pressure difference of 0.02 kg. per sq. cm. is required to close the contacts.

For d.c. welding operations, a series-connected contact relay is available. Similar relays are used for the supervision of electric motors. By equipping these contacts with an impulse organ of the Ferrari type, these contactors can be made to indicate a motor load. A modified type making contact only in the case of overload is also available. For the supervision of typewriters, the contact device shown in Fig. 7 is used.

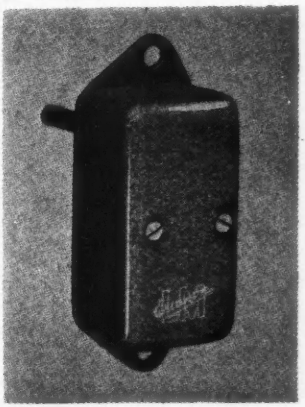


Fig. 7. **Typewriter contact device.**

AUTOMATIC LATHE WITHOUT CAMS

(From *Die Werkzeugmaschine*, Vol. 47, No. 6, March, 1943, pp. 133-134).

WHERE form tool work has to be done, the choice will lie between a manually operated turret lathe or an automatic operated by cams and working from the bar. Which of these machines is to be used will depend upon the number, shape, and dimensions of the components to be produced. Since the automatic requires special cams for every change in contour or size of the piece, special cams must be designed and made in each case. As this procedure is both costly and time consuming, the use of the cam-operated automatic is almost exclusively reserved for purposes of mass production, while series production is frequently carried out with the use of capstans.

The cam-less automatic shown in Figs. 1 and 2 represents a compromise between these two types of machines, combining their advantages but eliminating their shortcomings. It must therefore be considered as a very efficient machine for the automatic production in small series of screws and of shaped pieces. The machine can be used with either collet chucks for working from the bar, or with two- or three-jaw chucks,

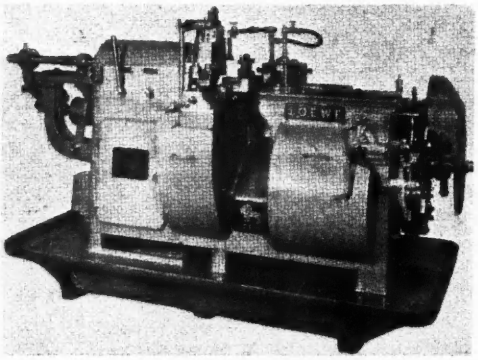


Fig. 1. **Cam-less automatic lathe.**

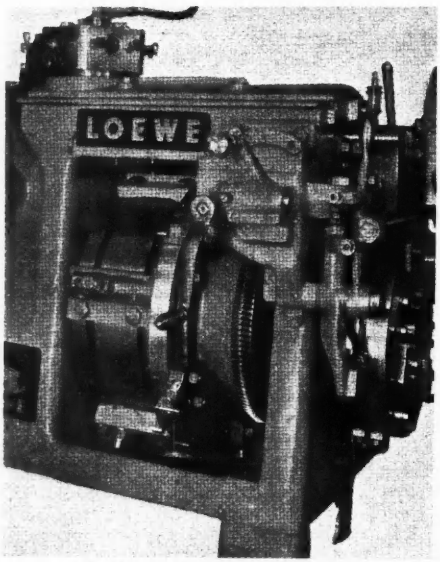


Fig. 2. **Operating mechanism of turret.** (Protecting cover removed).

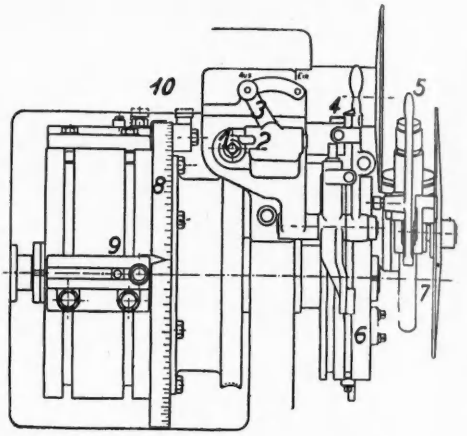


Fig. 3. **Detail of turret operating mechanism.** (1) Square end for attaching hand wheel ; (2) Safety lock ; (3) Lever for throwing in automatic feed ; (4) Lever for operating or idling position ; (5) Dial for feed rate with hand lever for fine adjustment ; (6) Indexing drum ; (7) Friction discs and gear rack with friction roller ; (8) Scale on drum effecting advance and return of turret ; (9) Slide with rollers for setting advance and return of turret ; (10) Rollers for effecting advance and return of turret.

these chucks being operated by hand or compressed air.

This machine has the advantage that it does not require the design of special cams for each individual work piece. Setting-up of the machine can therefore be effected in a minimum of time. The mechanism provided for operation of the turret is also worthy of note, its movements being produced by rollers (10) which are easily adjusted in proper position on the master drums (Fig. 3). The cross slides *a* (Fig. 4) are equipped with segmental pieces *b* clamped to discs provided with T-grooves for the purpose. The various feed rates are controlled by segmental pieces adjustably inserted in the grooves cut into the face of a master disc which revolves jointly with the master drum round a common shaft, the segments of the master disc positioning the wheel of a friction drive. The various adjustments are indicated by means of the dials (5) and (8) shown in Fig. 3. Once the necessary adjustments are made, a note may be made of the dial readings, so that the identical adjustment can be repeated on any future occasion.

The turret of the machine is made to hold five tools, of which one may be internally cooled by oil. The machine can be driven by an individual motor or from an overhead line-shaft. The headstock spindle is equipped for three speeds, i.e., one speed for turning and two for screw-cutting, the speed ratios being 1:2.5:7.5. The shaft which carries the master drum controlling the turret and the master disc adjusting the feed rates are driven from the headstock drive over a friction coupling, but it can also be driven direct so as to obtain rapid advance or withdrawal of tools between cutting operations.

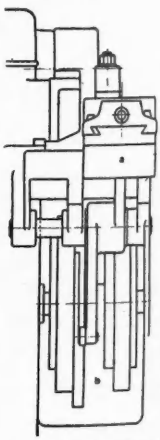


Fig. 4. **Outline view of cross slide, showing adjustable segments.** (a) cross slide ; (b) segment

THE INFLUENCE OF AIRSCREW THRUST ON TAKE-OFF DISTANCE.

By DIPL. ING. F. ROTH. (From *Flugwehr-und-Technik*, Vol. 6, Nos. 1 and 2, January and February, 1944, pp. 23-28 and 52-54).

IN estimating take-off distance, it has been customary in the past to replace the actual airscrew thrust by the static thrust for which some known value, e.g., 1.1 to 1.2 kg./h.p., was assumed.

The fact that it is now possible to calculate the static thrust with remarkable accuracy [11]* makes a reconsideration of the take-off problem desirable.

The take-off distance is measured from rest to that joint of the flight path at which, under steady climb conditions, the aircraft is to clear a 20 m. high obstacle (Fig. 1). It is subdivided into the following parts: the ground run proper, the transition arc and the climb proper.

The ground run is largely dependent on the coefficient of friction. This was dealt with in a previous article [10]. The equation of motion of the rolling aircraft (Fig. 2) is

$$P_{(v)} = M \frac{dV}{dt} = S - W - \mu\,(G - A) \qquad (1)$$

herein:

$P_{(v)}$	=	Accelerating force (being a function of speed) ..	kg.
M	=	$\frac{G}{g}$ = Mass of the aircraft	kg. sec.²/m.
G	=	Weight of aircraft	kg.
g	=	9.81 = Acceleration of gravity	m./sec.²
V	=	Speed of aircraft	m./sec.
t	=	Time	sec.
S	=	Airscrew thrust	kg.
W	=	Drag	kg.
μ	=	Coefficient of ground friction	
A	=	Lift	kg.
s	=	Take-off distance	m.

It is convenient to write eq. (1) as

$$P_{(v)} \cdot ds = M.d\frac{V^2}{2} \qquad .. \quad .. \quad .. \qquad (2)$$

which expresses the fact that the change in kinetic energy is equal to the work done. As $P_{(v)}$ is a complicated function of the speed, the equation is generally not integrated, but solved step by step. There is a great number of formulae for the ground run based on simplified assumptions, [1] [2] [3] [4] [5]. The author investigates the physical interpretation of these by starting with eq. (2) and substituting mean values (index m).

Thus, eq. (2) becomes

$$P_m \cdot s_r = \frac{M}{2} V_a^2 \qquad .. \quad .. \quad .. \qquad (3)$$

where V_a is the speed at unsticking.

Expressing P_m by mean values of thrust, drag and lift, after suitable transformation the ground run becomes

$$s_r = \frac{G/F}{\gamma \cdot Ca_a \left[\frac{S_m}{G} - \frac{1}{2}(\epsilon_a + \mu)\right]} \qquad .. \quad (4)$$

where, in addition to previous notations

F = Wing area .. .. m.²

$\gamma = \rho \cdot g$ = Specific weight of air kg./m.³

Ca_a = Lift coefficient at unsticking

$S_m = \frac{S_o + S_a}{2}$ Mean airscrew thrust $= \frac{1}{2}$ (static thrust + thrust at unstick)

ϵ_a = Drag to Lift ratio at unstick

Equation (4) is identical with that derived by H. Wenke [4]. For a given aircraft and its polare, s_r can be plotted against various parameters. This has been done, and is shown in Figs. 3-6. In calculating the take-off distance, the most uncertain factor is the transition arc, as the exact shape of the transition path depends very much on the way the aircraft is flown. This distance, however, is small when compared with the ground run and the climb proper, and can be neglected. This simplification is justifiable if, as is done by the author, the climb to 20 m. height is estimated from ground level (station a, Fig. 1), and will thus inherently include a part of the transition arc. The author shows that the transition arc $s_{ü}$ can be expressed easily so as to include that portion of the arc which corresponds to a height $h_{ü} = 10$ m. Then

$$s_{ü} = \sqrt{\frac{G}{F} \cdot \frac{4}{\gamma} \cdot \frac{h_{ü}}{\Delta Ca}} \qquad .. \quad .. \quad .. \qquad (5)$$

where the lift coefficient ΔC_a corresponds to the lift $\Delta A = A - G$.

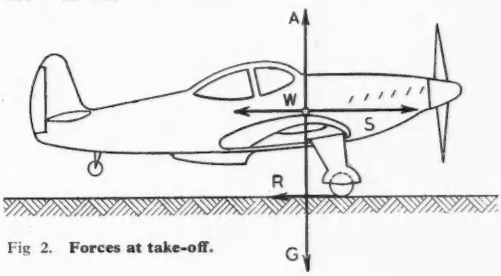


Fig. 2. **Forces at take-off.**

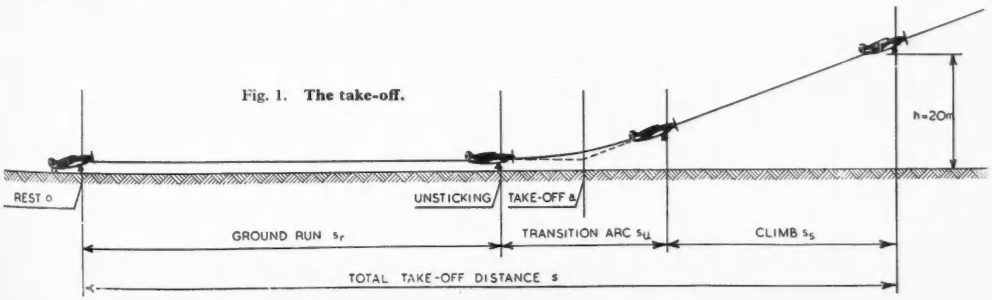


Fig. 1. **The take-off.**

* Numerals in brackets [ ] indicate the literature references at the end of the article.

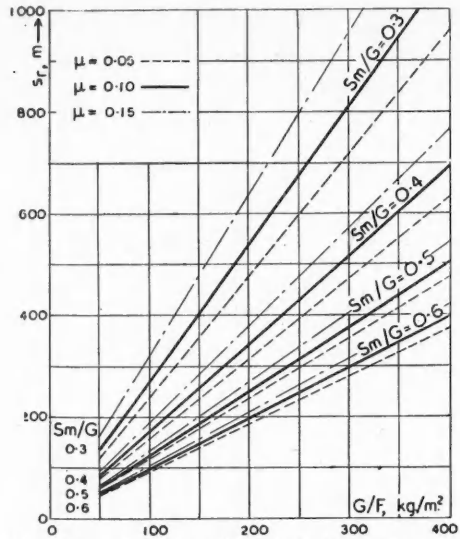


Fig. 3. **Ground run for various wing loadings with S_m/G as parameter.**
($Ca_a = 1.75$, $\epsilon_a = 0.175$, $\rho = 0.12$ kg. sec.²/m.⁴).

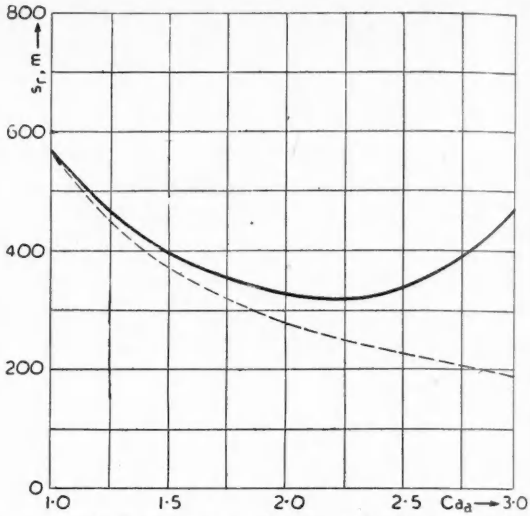


Fig. 5. **Ground run as a function of the lift coefficient at unsticking.**
(G/F = 200 kg./m² = 0.12 kg. sec.²/m⁴., $\mu = 0.1$, $S_m/G = 0.4$).
Dotted line: boundary layer suction at $\epsilon_a = 0.1$ const.

The eq. (5) is plotted in Fig. 7. References to this particular point are given in the literature [2] [7] and [8].

Bearing the above mentioned simplification in mind, the climb estimated from ground level is

$$s_s = \frac{h}{\frac{S_a}{G} - \epsilon_a} \quad .. \quad .. \quad .. \quad (6)$$

Fig. 8 shows the distance for climb under the stated conditions. Thus, the total take-off distance is obtained with sufficient accuracy from

$$s = s_r + s_s = \frac{G/F}{\gamma \cdot Ca_a \left[\frac{S_m}{G} - \frac{1}{2}(\mu + \epsilon_a)\right]} + \frac{h}{\frac{S_a}{G} - \epsilon_a} \quad (7)$$

To shorten the take-off distance, the wing loading $\frac{G}{F}$ should be as small as possible at large thrust values S_m and S_a. The development of modern aircraft brings about increasing wing loadings, which are the only means to increase the speed. Since an increase in lift coefficient beyond present values is rather limited (disregarding possible increase by means of boundary

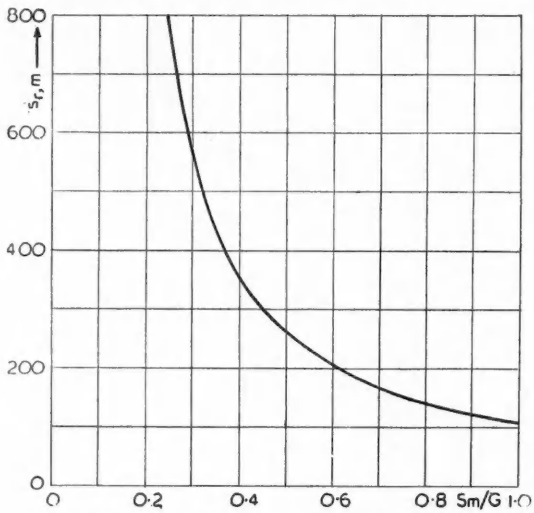


Fig. 4. **Ground run as a function of S_m/G.**
(G/F = 200 kg./m.², $Ca_a = 1.75$, $\epsilon_a = 0.157$, $\mu = 0.1$, $\rho = 0.12$ kg. sec.²/m.⁴).

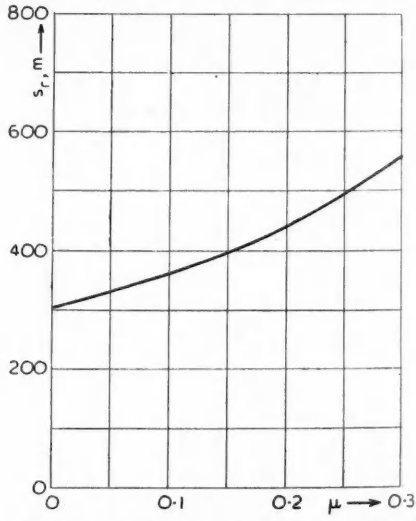


Fig. 6. **Ground run as a function of the coefficient of friction.**
(G/F = 200 kg./m.², $Ca_a = 1.75$, $\epsilon_a = 0.157$, $S_m/G = 0.4$, $\rho = 0.12$ kg. sec.²/m.⁴).

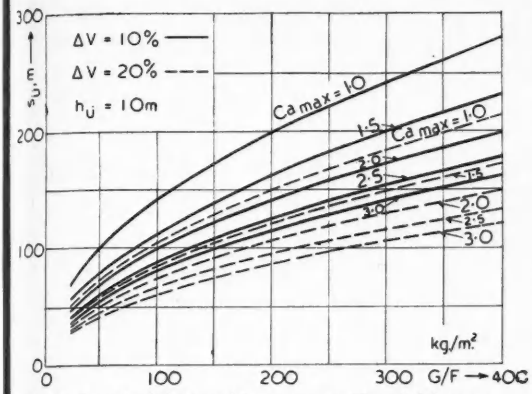


Fig. 7. **Length of the transition arc up to 10 m. height as a function of wing loading.**

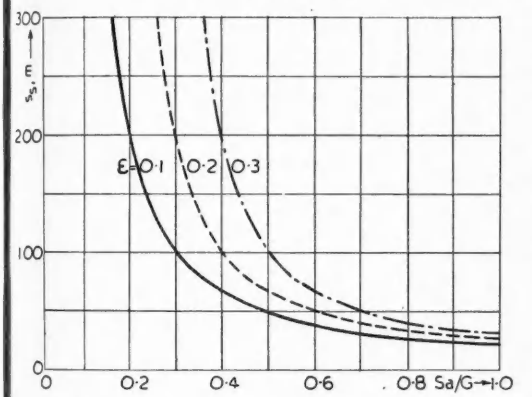


Fig. 8. **Distance for climb up to h = 20 m. against** Sa/G.

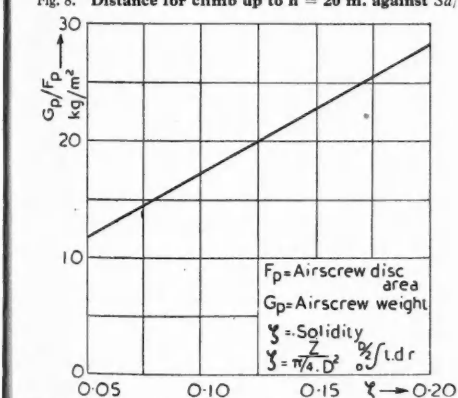


Fig. 9. **Specific airscrew weight plotted against solidity.** F_p = Airscrew disc area. G_p = Airscrew weight. ζ = Solidity.

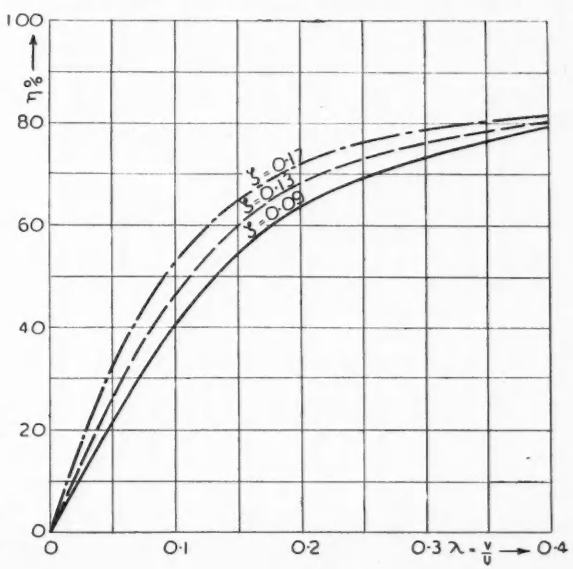


Fig. 10 (right). **Airscrew efficiency against advance ratio for three values of solidity.**

(Power N = 1300 h.p., diameter D = 3.3 m., peripheral speed U = 260 m./sec. air density, ρ = 0.12 kg. sec.²/m.⁴).

layer control) an increase in the thrust value offers the solution in reducing the take-off distance.

Means to increase the static thrust have been discussed [11], and will be only briefly reviewed. It is possible by increasing the total blade area/disc area within practical limits, to increase the static thrust by 30 per cent. By suitable choice of reduction gear a reduced r.p.m. for maximum speed will permit the use of a greater airscrew diameter. This, and a short duration increase in r.p.m.'s during take-off make an increase in static thrust possible of as much as 80 per cent. Clearly an increase in solidity and diameter affects the airscrew weight, and thus the wing loading. This is shown in Fig. 9, and it can be seen that an increase in solidity from the normal value $\zeta = 0.08$ to 0.10 to $\zeta = 0.15$ to 0.20 is accompanied by an increase in airscrew weight of 50 to 60 per cent. The weight of the airscrew is about 2 to 5 per cent of the total weight of the aircraft, and thus, even a doubling of the airscrew weight does not increase the wing loading by more than 5 per cent.

No doubt, this increase is well repaid by a 50 per cent increase in static thrust.

The increase in static thrust is, however, not the sole measure for improvement in take-off. Eq. (7) requires large values for S_m and S_a. These values are also increased with increasing ζ, and so is the climb performance as Fig. 10 shows. The three airscrews shown in the figure have been tested in wind tunnels, and had a maximum efficiency of $\eta = 85$ per cent at the advance ratio $\lambda = 0.6$.

The author has estimated the changes in take-off distance for a number of standard aircraft. The range for wing loading was 160 to 306 kg./m.², that for power loading 2.3 to 4.5 kg./h.p. These examples show that a take-off distance of not more than 500 m. is possible, even with an aircraft of both high wing-loading and high power-loading. In the most unfavourable case 720 m. were required, a distance not exceeding present day airfield dimensions.

For the future, these conclusions are of importance, inasmuch as present planning of post-war aerodromes appears to put aside unnecessarily large capital sums for extension of runways, whereas this capital should be used for development of efficient aircraft on the line shown in this paper. The increase in wing loading

would lead, at the same time, to economic aircraft for long distances.

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MACHINE FOR SYNCHRONIZED, COMBINED BENDING AND TORSIONAL FATIGUE TESTS.

By Dr. ING. E. BRUDER. (From *Zeitschrift des VDI*, Vol. 87, No. 5/6, 6th February, 1943, p. 82.)

BENDING and torsional fatigue tests of crankshaft steels and crankshaft parts on conventional testing machines do not produce results which could be regarded as simulating actual conditions. To bridge the gap, a testing machine has been developed which allows synchronized loading in both bending and torsion in conformity with actual conditions.

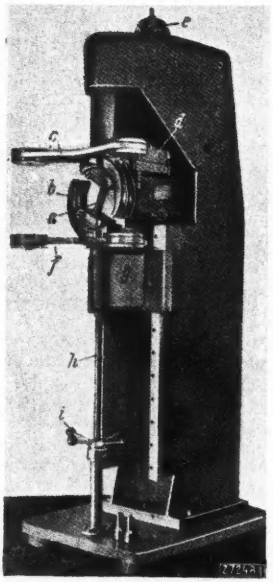
With the testing machine shown in Fig. 1, the ratio between direct and shear stress δ/τ can be chosen between zero and infinite, according to an angle of position α of 0 deg. and 90 deg. of the test piece.

The essential part of the machine is an apparatus shown in Fig. 2 which is mounted between the fixing heads of a torsional fatigue testing machine with resonance drive. The bending moment M_b and the torque M_d on the test piece are obtained from the torque of the machine M by

$$M_b = M \sin \alpha$$
$$M_d = M \cos \alpha$$

The test piece remains stationary during testing so that the procedure of rupture can be observed exactly. Thus, with cylindrical test pieces the maximum stresses always recur at the same spot, in two diametrically opposed points.

The apparatus has been built for test pieces of 30 mm. diameter. The resonance machine used for testing allows the application of a static pre-stress of arbitrary magnitude.



a Test piece.
b Apparatus for fixing the test piece (compare Fig. 2).
c Upper swing, driven by out-of-balance-weight *d*.
d Out-of-balance-weight driven by electric motor *e*.
e Electric motor.
f Lower swing fixed on support *g*.
g Support, adjustable by screw *h* and hand wheel *i*.

Fig. 1.
Machine for bending and torsional fatigue tests.

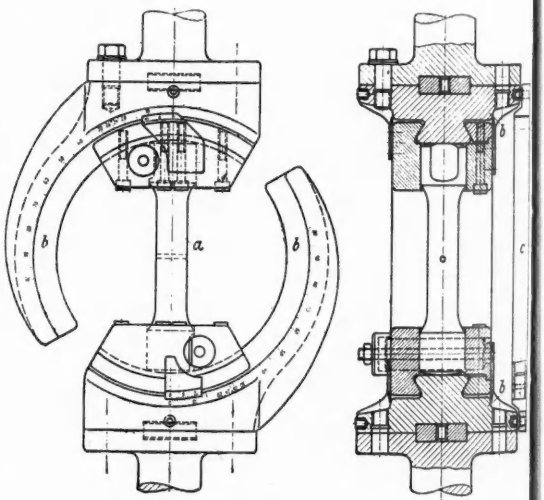


Fig. 2 **Apparatus for fixing the test piece.**
a Test piece, 30 mm. diameter.
b Loading device with angular scale for positioning of test piece.
c Centering device.

SHOCK ABSORBING SUSPENSION AND SHOCK ABSORBERS.

By M. FORESTIER and M. LEHEUNE. (From *Revue Générale des Chemins de Fer*, Vol. 63, No. 1, January/February, 1944, pp. 1-10).

IN the most commonly used spring suspensions the spring deflection is approximately proportioned to the loading applied, with the frequency of oscillation remaining constant and independent of the amplitude of deflection. A drawback of spring and suspension systems of this type is the absence of a damping effect when the state of equilibrium has been disturbed; while resonance may occur if the natural frequency of the spring happens to be coincident with the frequency of the disturbing force. In order to avoid the occurrence of resonance various methods have been evolved to impart suitable characteristics to spring supports, without having recourse to solid or liquid damping devices. Arrangements of this kind can be grouped as follows:

(1) Springs showing progressively diminishing resilience when the load is increased or decreased relative to its specified equilibrium value.

(2) Springs of constant resilience equipped with a mechanism which produces a non-proportional deflection, the deflection usually being made to increase progressively with the load.

In group (1) the following suspension devices may be listed:

(1) *Helical springs* with varying section of the coil. By making the end coils comparatively weak and correspondingly flexible, a small load will suffice to bring these end coils into mutual contact, so that further compression of the spring takes place by deflection of the stiffer intermediate coils. This system as applied to automobile suspension springs is exemplified by the Beck Suspension indicated in Fig. 1.

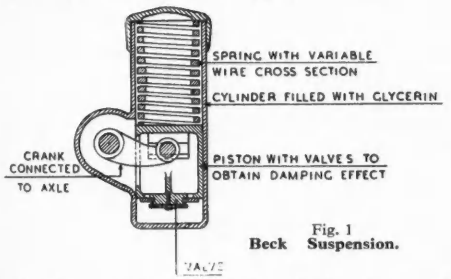

SPRING WITH VARIABLE WIRE CROSS SECTION
CYLINDER FILLED WITH GLYCERIN
PISTON WITH VALVES TO OBTAIN DAMPING EFFECT
CRANK CONNECTED TO AXLE
VALVE

Fig. 1
Beck Suspension.

(2) *Volute type* or conical helical springs with diminishing deflection under increasing load. This characteristic will be accentuated if the large diameter coils forming the outer portion of the spring possess a progressively decreasing thickness.

(3) *The spring suspension suggested by Leboucher.* Referring to Fig. 2, the load A rests on a device composed of a number of ordinary helical springs R_1 surrounding a conical helical spring R_2. The upper conical helical spring R_3 will be depressed whenever an oscillation depresses the main group of springs R_1 below its position of static equilibrium. In this case, spring R_2 is prevented from further compressing when the lower circular edge of the plunger supporting R_3 has descended upon its seat which is formed by the topmost rim of bore B. The deflection versus load characteristic of this spring assembly is given in Fig. 3.

Devices belonging to group (2) appear to have found only limited employment, and up to the present have only been used for light or very light vehicles such as motorcycles and perambulators.

Summarising, it may be said that the devices described above are not subject to resonance if the frequency of cyclic load variation is constant. They are not, however, entirely free from pseudo-resonance

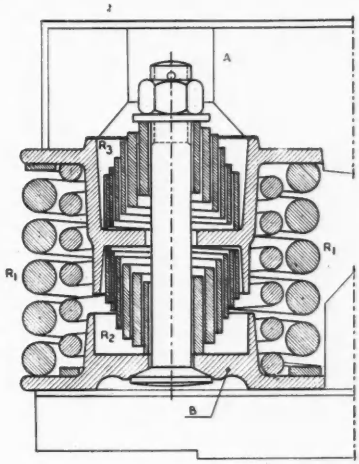


Fig. 2. **Leboucher Suspension.**

effects which may occur if the oscillations are sufficiently large. Furthermore, Leboucher's device requires perfect adjustment of the height of the springs and is ill adapted to loading in excess of the normal value, as e.g., overloading of a vehicle. As a matter of fact, if the device is overloaded, the point of static balance will be removed from the point of inflexion C (Fig. 3) to a location on the upper branch of the curve. This will have a profound effect upon the operating characteristics of the device and will rob it of its original advantage which consists in its high degree of resiliency under normal load conditions. Also, these devices are subject to wear, being used as a direct mechanical link for load transmission.

SHOCK ABSORBERS.

Vibration damping and shock absorbing characteristics are imparted to suspension devices by the incorporation of solid or liquid damping devices which oppose the production of large oscillations induced by resonance. The damping effect may be obtained in several ways, such as with the exclusive use of springs

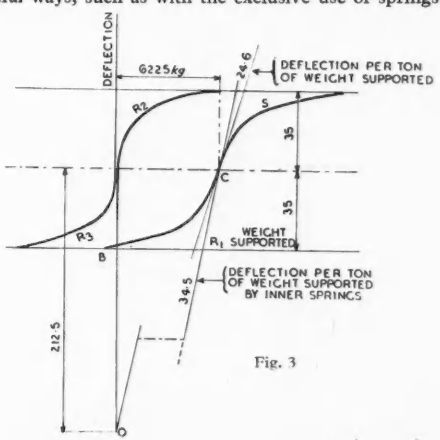


Fig. 3

with inherent friction effect (laminated springs); or a combination of frictionless springs with friction-effect springs may be employed; or independent damping devices may be fitted to the spring suspension concerned.

To a certain degree laminated spring suspensions are capable of counteracting large oscillations; but more often than not their damping effect is insufficient because of the greasing of the individual leaves, which may also be highly polished. This applies in particular to passenger automobiles. On the other hand, in the case of railroad carriages, the slightly rough surface of the leaves and the absence of lubrication may provide excessive damping. Moreover, under such conditions, deformation of the spring will only take place whenever the imposed loading exceeds the static frictional resistance. Thus the spring will transmit a certain amount of vibration which considerably affects the comfort of travel.

Referring to the matter of automobile springs which require an increase in the friction between the component leaves, in one design this is achieved by adjusting

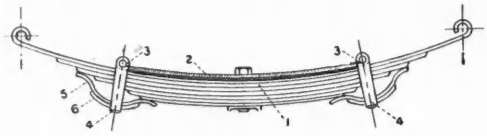


Fig. 4. **Shock Absorber Fram-Vermot.**
1 Ordinary leaves, 2 Counter leaves. 3 Attachment of counter leaves, 4 Hooks, 5 and 6 Semi-elliptical auxiliary springs.

the pressure between the leaves in the manner shown in Fig. 4. A still more simple procedure consists in bringing the damping effect into play at the moment of the rebound. To be effective, these devices require the occurrence of large oscillations, and it is for this reason that they hold little promise with regard to their application to rolling stock. Definitely, the leaf type spring, when used by itself is, more often than not, unsatisfactory both as spring and as damping device.

COMBINED FRICTION TYPE AND FRICTIONLESS SPRING DEVICES.

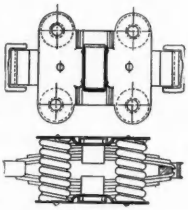


Fig. 5

Devices of this kind have apparently found some favour in American practice, where they are used for the bogies of certain types of railroad trucks. A typical device of this kind is illustrated in Fig. 5. In this design the laminated spring acts both as spring and as damper. The magnitude of the damping effect will depend upon the respective resiliencies of the component springs. However simple these types may be it would appear preferable to separate the functions of resilience and damping.

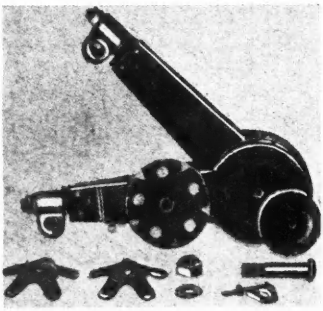


Fig. 6. **Shock Absorber Hartford.**

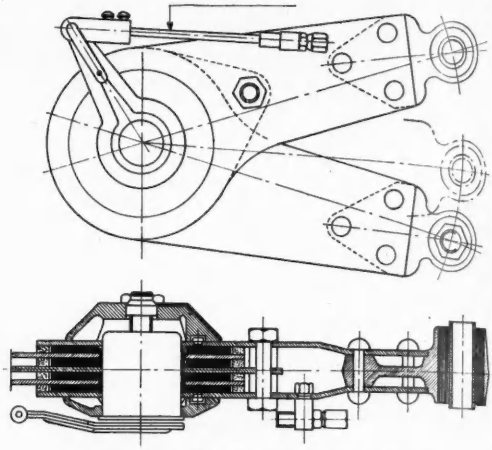


Fig. 7. **Shock Absorber Repusseau-Silentbloc.**

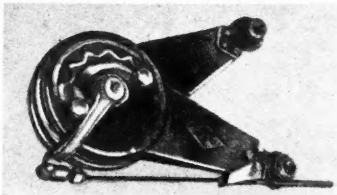


Fig. 8. **Shock Absorber Apex.**

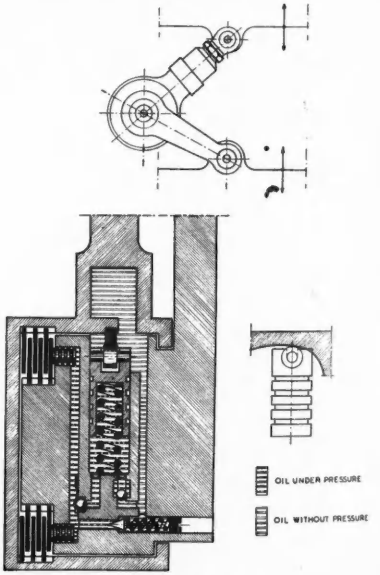


Fig. 9. **Shock Absorber Ram.**

SHOCK ABSORBERS OR DAMPERS PROPER.

(a) *Solid-friction type devices.* These may be either of the constant or of the progressive damping variety :

(1) *The solid-friction type shock absorbers with constant damping effect.* With this type damping is unaffected by the nature of the imposed vibrations. The damping effect itself is achieved by the use of two concentric and rotating rubbing surfaces, the required surface pressure being produced by suitable means. The two halves of the device are carried on levers, one of which is connected to the body of the vehicle and the other to the axle. A typical example is the Hartford Shock Absorber shown in Fig. 6. The pressure of the rubbing surfaces can be adjusted when the vehicle is at rest, or a remote control may be fitted to permit adjustment while the vehicle is in motion. Examples of the latter type are the Repusseau-Silentbloc and the Apex Shock Absorbers illustrated in Figs. 7 and 8 respectively.

Solid friction type shock absorbers with progressive damping effect. A representative example of this class is the Ram Shock Absorber illustrated in Fig. 9. In this design the pressure between a number of friction-discs is produced by the oil pressure set up by a small pump with cam-operated plunger. The stroke of this plunger, and with it the oil pressure, depends upon the relative movement of the parts between which the device is interposed. The attractiveness of this design, however, is more apparent than real, considering the complexity of its mechanism.

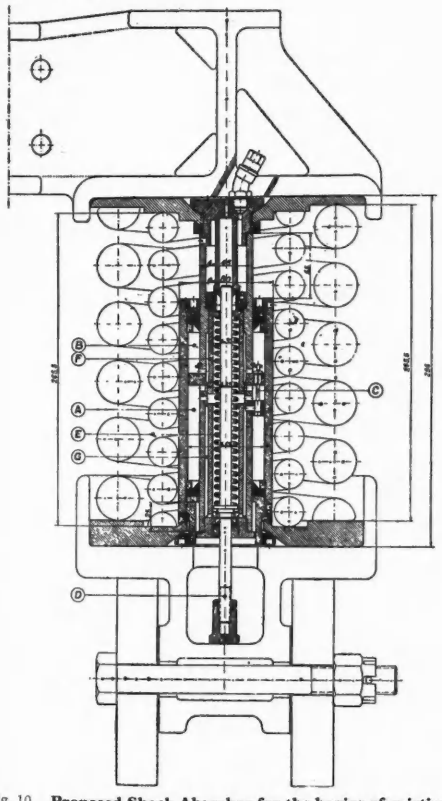


Fig. 10. **Proposed Shock Absorber for the bogies of existing steel-built RR carriages of the S.N.C.F.**

HYDRAULIC SHOCK ABSORBERS.

In the majority of hydraulic shock absorbers, the friction effect is obtained by forcing a liquid to pass through an orifice or through very narrow channels. According to the type of flow resistance and the viscosity of the liquid employed, the friction effect will vary as the square of the velocity or in direct proportion with the latter. Nearly all well-known designs embody the former principle. Representative makes of this type are the Houdaille shock absorber of the chamber type and the Repusseau and the Broulhiet shock absorbers, the two latter incorporating pistons.

In view of certain shortcomings encountered with the aforementioned types, the Soc. Olaer-Marine has proposed a design in which the frictional resistance is obtained with the use of a spring-loaded valve with large flow area. In principle this device consists of a piston connected to the one moving part and a cylinder connected to the other. The flow restraining valve is located in the piston, the valve spring being loaded in proportion with the amplitude of the piston travel. Thus the counterpressure acting upon the piston is proportional to the elongation of the supporting springs. But this action of the shock absorber only takes place when the springs expand and not when they are compressed.

Two different designs of this shock absorber are shown in Figs. 10 and 11. Here it will be seen that during the compression stroke the oil must flow from reservoir *A* to reservoir *B* via the check-valve *C*, the spring *E* being compressed by the spindle *D*. But when the supporting springs expand, the oil in reservoir *B* can return to reservoir *A* only by lifting valve *F* which is loaded by spring *E*. The tension of the latter spring remains unchanged during the expansion stroke, as check valve *C* is now closed, thus shutting off the reservoir *G*. This type of shock absorber ensures a strictly aperiodical characteristic. A further advantage is that its action is practically independent of changes in oil viscosity with the temperature.

The second part of the paper presents the mathematical theory underlying the various types of damping devices, and indicates in each case the extent to which practical application is possible.

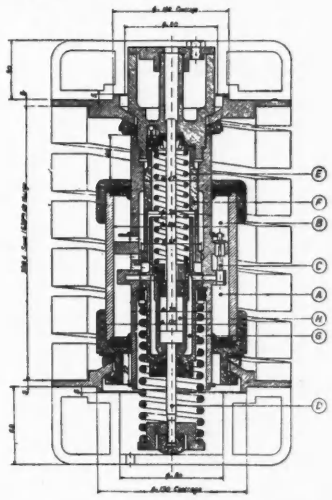


Fig. 11. **Proposed Shock Absorber for Swiss RR carriages and also for French RR carriages designed by Usine de Schlieren.**

POWER-RECOVERY FROM TESTING AIRCRAFT ENGINES.

By R. MODLINGER. (From *E.T.Z. Elektrotechnische Zeitschrift*, Vol. 65, Nos. 7/10, March 9th, 1944, pp. 73-76).

THE problem of utilisation of electrical energy although investigated a few years ago, was again taken up following measures of economy introduced by the Minister of Armaments and Supply.

Originally, the braking of aircraft engines was carried out by means of Leonard converter sets. This problem has also received increased attention in the U.S.A. and in England. In the U.S.A. since 1942 they have used synchronous generators working in conjunction with a hydraulic slip-coupling of the Voith-Sinclair type, whereas in England Leonard converter sets are in general use with these brakes.

The author enumerates six different methods of brake tests, which are shortly described as follows :—

(1) Braking by means of a d.c. machine and the return of energy to the network of electric power supply by means of a Leonard converter set. The range of speed regulation is very wide, but the efficiency, especially at light loads, is rather poor and was due to losses in three sets of machines. Other disadvantages were high capital cost and additional space requirements for the converter set and switchgear plant.

(2) Braking by means of a d.c. machine and the return of electrical energy by means of a single armature d.c.-a.c. invert converter. The remarks made under item (1) also apply to this method, and the efficiency is not improved by the substitution of an invert converter set for the Leonard set.

(3) Braking by means of a synchronous generator working in conjunction with a Voith-Sinclair coupling. By this method the electrical braking can only be continued near to, but not below synchronous speed of the a.c. machine. The losses at the coupling consist of the slip loss, determined by the slip revolutions, the torque transmitted, and the friction losses. At a slip of about 3-5 per cent the whole torque can be transmitted.

The electric machines are built as 6-pole or 8-pole machines, according to the type of aircraft engine to be tested and for an average slip of 4 per cent the corresponding speeds, at or above which electric braking may take place, is limited to 780 r.p.m. and 1040 r.p.m. respectively. In the range of speeds below these limits braking must be arranged by means of an additional propeller fixed to the shaft, or by means of a hydraulic brake. This part of the energy cannot be utilised.

It was proposed to effect braking by means of a Voith-Sinclair coupling by blocking the rotor of the synchronous machine. If this could be designed of suitable dimensions it would form a simple way to carry out braking at these lower ranges of speed.

(4) Braking by means of an asynchronous generator with slip rings. As regards operation and losses this method corresponds with the method under item (3), with the advantage of eliminating the use of a hydraulic coupling. On the other hand, resistances become necessary in the rotor circuit. The energy dissipated in this way may be utilised for heating purposes.

(5) Braking by means of a three-phase commutator machine. With such a machine, or a suitable 3-phase regulation set, it is possible to brake at any desired speed, similar to the d.c. pendulum machines. The motors, however, require an existing 3-phase supply for the magnetisation. A stable frequency of this network is also needed for the stabilisation of the brake sets.

(6) Braking by means of hydraulic pumps and the recovery of energy similar to a pump storage installation, or hydraulic solutions. This method becomes rather complicated because of the necessity to provide separate pressure-water mains for each test stand. In the absence of separate pipes, the brake energy produced at different speeds of the several aircraft engines and tested simultaneously, will cause the formation of whirls in the pipe, and the energy to be dissipated in heating the water. It is further possible that whilst working at the lower ranges of speed, the pumps will produce too small a head to enable any flow of water through the pipeline.

The capital costs of this method are considerable and the site must be suitable for the erection of a fairly large storage reservoir. For aircraft plants of the usual size, the capacity of such a reservoir should be about 1,000,000 cu.m. with a head of 35 m. It will take a lengthy period to build such an installation.

BRAKING METHODS FOR AIRCRAFT ENGINES.

Fig. 1 is a diagrammatic illustration of the component parts of the testing plant used in the various methods. Table I. gives the losses incurred when recovering energy through braking of aircraft engines.

As the method under item 2 is nearly identical with method 1, and the method 6 is scarcely a practical proposition, only the other four methods are further considered.

Regarding the range of speed, methods 1 and 5 offer practically the same advantages and in this respect methods 3 and 4 are of equal value. Fig. 2 illustrates the amount of recovered energy from a type

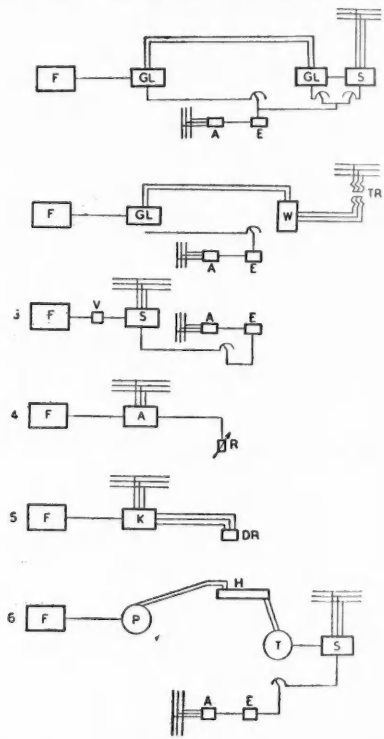


Fig. 1

F = Aircraft engine. *GL* = d.c. machine. *S* = Synchronous machine. *A* = Asynchronous machine. *W* = Rotary converter. *TR* = Transformer. *E* = Exciter. *R* = Resistances. *K* = Commutator machine. *DR* = Regulating set. *P* = Pump. *H* = Storage tank or reservoir. *T* = Turbine. *V* = Voith-Sinclair coupling.

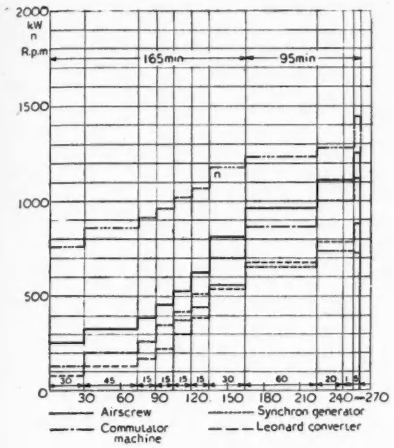


Fig. 2.
Time of run, r.p.m., and output of aircraft engine, Type BMW 801.

TABLE I.

Methods.	Losses.
1. Leonard converter.	Gl + U + Gl + S + EU
2. Rotary invert converter.	Gl + U + W + T + EU
3. Synchronous machine with hydraulic coupling.	VS (Sl + Frict.) + S + EU
4. Asynchronous machine.	A (Sl + a
5. Commutator machine.	K (Dr + k)
6. Pumping plant.	P + L + Tu + S + EU

Gl = Losses in d.c. machines. *U* = Losses in electric cables *S* = Losses in synchronous machine. *EU* = Losses in exciter converter. *W* = Losses in Rotary converter. *T* = Losses in transformer. *VS* = Losses in turbo-coupling. (*Sl* = Slip losses, Frict. = Friction loss). *A* = Losses in asynchronous machine. (*Sl* = Slip loss, a = Stator loss, and rotor loss). *K* = Losses in commutator machine. (*DR* = Regulating set, k = Stator and rotor losses). *P* = Losses in pump. *L* = Losses in pressure pipe. *Tu* = Losses in turbine.

BMW 801 aircraft engine. In the curves the friction losses in the couplings (not in the Voith-Sinclair turbo-coupling) and the electrical losses in the exciter sets and cables are omitted. The energy regainable by method 4 is not indicated as it does not differ much from the value shown by the curves of method 3.

Should it be necessary to use with method 3 an additional propeller, this will result in producing air-eddies and consequently more losses, about 100 kW. at 1450 r.p.m., but this figure has not been deducted from the energy figures of method 3, as it may be possible to dispense with this second propeller within the lower ranges of speed, by blocking the rotor of the synchronous machine and using the Voith-Sinclair coupling as a hydraulic brake.

Method 1 is second best with regard to recovery of energy, although it involves high capital costs. The highest efficiency is obtained with method 5 which shares all the other advantages with system 1.

Under present conditions of working, about 180 minutes are needed for erecting and dismantling an aircraft engine test stand. According to Table II. the time taken for running-in and the preliminary run amounts to 260 mins., so that the total time including erecting and dismantling amounts to 620 mins.

Assuming two working shifts per day, one test stand can deal with

$$\frac{450 \times 60}{620} = 43 \text{ aircraft engines per month.}$$

According to the figures of Table II. the recovery by means of commutator machines (method 5) would be .. .. .. 104,200 kWhr. per month corresponding to an annual recovery of 1,250,000 kWhr.

By means of method 3 the figures for recovery per month would amount to .. .. 68,500 kWhr. or an annual recovery of .. .. 821,000 „

It is contemplated in the future to reduce the time taken for the erection of test stands, the running-in and preliminary running times being also cut down, so that the total time should be reduced to 275 mins.

With each test stand one could then handle

$$\frac{450 \times 60}{275} = 98 \text{ aircraft engines per month.}$$

The electrical energy recoverable on each test stand would then be :

By means of method 5	143,500 kWhr.	per month
	1,723,000 „	annually
By means of method 3	107,600 „	per month
	1,290,000 „	annually

With the introduction of electrical braking systems in aircraft engine factories, it will be possible to generate about 80 per cent of the energy required for the operation of the whole of the works.

The setting of a definite speed-torque characteristic does not involve any difficulties in either of the two methods mentioned above.

In an aircraft engine factory with an output of 2000 engines per month with the utilisation of recoverable electrical energy from the test stands the saving would be $0.7 \times 2000 \times 12 \times 2424 = 40{,}730{,}000$ kg. coal per annum, equal to a load of about 2700 waggons, or more than 65 trains fully loaded.

To protect the aircraft engines as well as the brake machinery, they should be brought to a standstill by means of a centrifugal switch, when disconnecting the brake generators from the network.

TABLE II.

Recovery of Energy from Run-in and Preliminary Runs of Aircraft Engine, Type BMW 801.

Time Min.	R.p.m. Propeller	Output h.p. Propeller	Leonard %	Leonard kW.	Leonard kW. hr.	Synchr. Gen. %	Synchr. Gen. kW.	Synchr. Gen. kW. hr.	Commutator machine. %	Commutator machine. kW.	Commutator machine. kW. hr.
30	752	350	0.32	83	40	—	—	—	0.5	130	65
45	860	450	0.41	135	101	—	—	—	0.62	205	156
15	913	530	0.45	175	44	—	—	—	0.68	265	66
15	967	625	0.50	230	57	—	—	—	0.75	345	86
15	1020	725	0.565	300	75	0,73	383	95	0.79	425	106
15	1073	850	0.62	390	97	0,78	455	114	0.83	520	130
30	1182	1100	0.675	550	275	0,83	570	285	0.87	710	355
60	1235	1310	0.70	680	680	0,85	664	664	0.895	870	870
30	1288	1510	0.71	790	395	0,86	743	372	0.90	1000	500
5	1450	1710	0.70	880	73	0,85	737	61	0.89	1120	93
260	—	—	—	—	1837	—	—	1591	—	—	2424
95	—	—	—	—	1148	—	—	1097	—	—	1463

The running-in and preliminary run test stands could be considerably simplified in equipment and reduced in size. The output of the engines might be ascertained with sufficient accuracy by the readings of electric measuring instruments, a special torque balance may be omitted.

Taking into consideration the size of the existing power station in the works, one may equip 10-20 per cent of the test stand units with synchronous generators which can be utilised for supplying current in cases of emergency. The remainder of the test stands should be equipped with commutator machines, or for the sake of simplicity in operation, with asynchronous machines. The excitation of all machines should be synchronised, so that each generator should carry the same idling load. As the machines will not carry the same effective load care must be observed for the stabilisation. In such a way the synchronous machines take care of the supply of magnetising current for the whole network. The commutator machines may, if desired, be then called upon to supply the inductive idling load.

The maintenance of the frequency in the disconnected network is achieved by a special quick-acting load-regulator.

As the available energy will most likely be insufficient to feed the whole plant, overloading in the network should be avoided by a timely reduction of the load, as otherwise a drop in the frequency cannot be avoided.

No special problems are involved in the selection of switchgear plant for test stands equipped with commutator machines, asynchronous or synchronous machines.

The amount of constructional material is largest for test stands with commutator machines and smallest with asynchronous machines. As the commutator machines yield the greatest economy, it will be the best policy to equip a large number of test stands with such types of machinery.

OXYGEN-PRESSURE-GASIFICATION OF SOLID FUELS.

By W. OFFENBERG. (From *Stahl & Eisen*, Vol. 63, No. 51, December 23rd, 1943, pp. 936-939).

THE present-day state of German town gas production is characterised by the existing equilibrium between the gas demand and the available gas production of existing coking plants. However, the increasing demands for gas by the steadily growing gas-consuming industries call for the opening up of new sources of gas, thus compelling the German gas industry to develop novel methods for the complete gasification of coal.

The principle of the pressure gasification process is outlined in Fig. 1. Its oxygen requirements are met by an oxygen production plant of the Linde-Fränkel type which, during the last few years, has found very large scale application in industrial oxygen production by decomposition of air. Oxygen is compressed to 20-30 kg. per sq. cm. Steam of the same pressure, but superheated to 500 deg. C. is also required. The oxygen and the steam are passed into the gas generator in which gasification proceeds at 20-30 kg. per sq. cm. pressure. The raw gas is then passed into a condensing plant in which the tar, oil, and the water vapour contained in the gas are fractionally condensed.

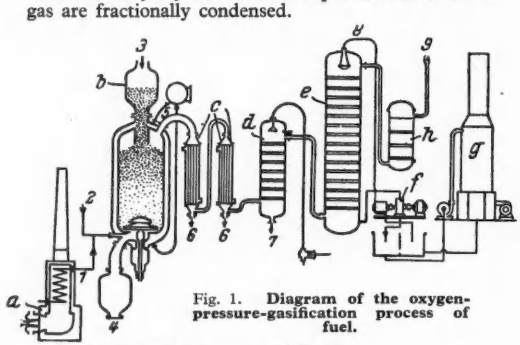


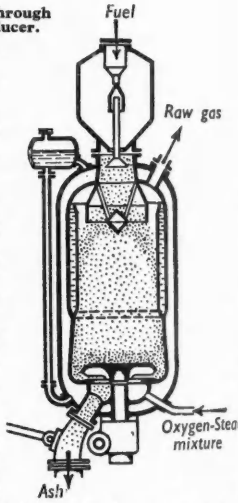
Fig. 1. **Diagram of the oxygen-pressure-gasification process of fuel.**

(a) Steam superheater
(b) Pressure gas produced
(c) Cooler
(d) Oil scrubber
(e) Pressure water scrubber
(f) Pump-turbine
(g) Aerating tower
(h) Sulphur removal
(1) Steam
(2) Oxygen
(3) Fuel
(4) Ash
(5) Dirty gas
(6) Tar and oil
(7) Benzene
(8) Circulating water
(9) Clean gas.

The temperature of the raw gas at the outlet of the gas producer amounts to 300-600 deg. C. according to the moisture content of the fuels, and decreases to 20-30 deg. C. upon passing through the coolers. The benzene is subsequently absorbed by wash oil in a scrubbing tower and the extraction of the absorbed benzene is effected by distillation at atmospheric pressure, the wash oil being subsequently recirculated into the scrubbing tower by means of a high-pressure pump.

The gases are then freed from carbon dioxide and most of their sulphuretted hydrogen contents by pressure washing. The scrubbing water is circulated through the scrubber by means of a high pressure pump and upon the discharge from the scrubbing tower passed through a turbine which it leaves at atmospheric pressure. It is thereupon regenerated by aerating it in a tower from whence it is returned to the suction side of the circulating pump. The process of gas purification is concluded in a dry-type scrubber with a bog-ore filling. At this point the gas pressure still amounts to some 20-25 kg. per sq. cm. which is sufficiently high to transmit the gas to distant points of consumption without requiring any further compression.

Fig. 2. **Section through pressure-gas producer.**



The latest type of gas producer, shown in Fig. 2, consists of a refractory lined pressure vessel which for reasons of safety is provided with a cooling water jacket. At the top there is a sluicing device for the continuous coal feed, another sluice for ash removal being provided at the bottom. The incandescent fuel bed of this gas producer is carried on the usual type of rotary grate, the speed of the latter being determined by the amount of slag discharged.

This method of high pressure gasification possesses a number of distinctive features as compared with the older methods of gas production. The reaction products which are formed in the gasification of a solid fuel with the use of an oxygen steam mixture are carbon monoxide, hydrogen, carbon dioxide and methane, a portion of the steam remaining undecomposed. The relative amounts of these products of reaction are largely conditioned by pressure, temperature, and time of reaction (Fig. 3). The composition of the gas produced can be adjusted within wide limites by controlling the process pressure, the amount of steam added, and the specific loading of the producer shaft. Where town gas

is to be produced, the process it adjusted to favour the production of methane and carbon dioxide in addition to carbon monoxide and hydrogen.

This method is the first to make it possible to utilise the methane synthesis in the gasification process. It increases the calorific value of the gas to such an extent that after the removal of the carbon dioxide the product is equivalent to town gas, thereby realising the long sought-after complete gasification of solid fuel.

A first plant of this type was installed by the A. G. Sächsische Werke in 1936, its annual output being 3.5 million cu. m. of town gas transmitted under its original pressure to a distant town. The efficiency of this plant is 85.2 per cent. The satisfactory operating results obtained led to the building of a larger plant with an initial capacity of 80 million N.P.T. cu. m. of gas per year. Since then the plant has been doubled in capacity.

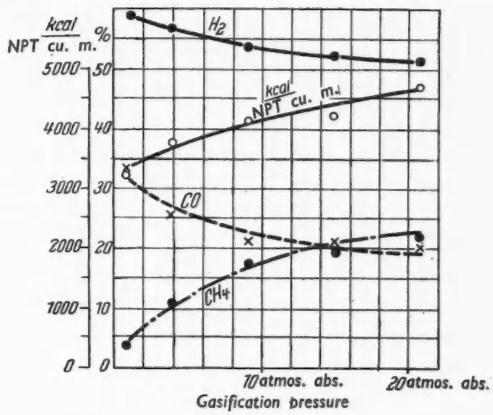


Fig. 3. **Influence of gasification pressure upon the composition of the gas.**

HOW TO CHOOSE AND USE CARBIDE-TIPPED TOOLS.

By J. JACQUET. (From *La Pratique des Industries Mécaniques*, Vol. 26, No. 10, January, 1944, pp. 147-152).

GENERAL RECOMMENDATIONS.

1. Use the quality of carbide-tip most suitable for the work to be done. Every manufacturer has several grades of carbides to suit metals of different hardness, so precise details should be given when ordering.

2. For the body of the tool use a tough steel, easy to braze and not liable to be spoilt or distorted during heating up to 1100/1150 deg. C. Semi-hard carbon steel is about the best.

3. Fix the tool with the minimum overhang.

4. Avoid taking heavy cuts at low speeds. It is better to increase the speed than the cut.

5. Never engage the tool before the workpiece is rotating at cutting speed.

6. Never stop while cutting. Withdraw the tool fully before stopping or going back.

7. See that lubrication, where required, is abundant and continuous.

8. Use the tools carefully, without clashing against the workpiece or banging with a hammer or tool to adjust them.

9. Do not leave tools about on the benches or machine tool beds. If possible, protect cutting edges with a wooden cover to avoid knocks.

10. Cutting edges should always be kept sharp by frequent grinding by a stone or diamond dust bar.

11. Grinding should only be done on suitable wheels and by a very careful specialist in grinding.

12. Avoid excessive heat when grinding. Do not cool by plunging the tool in water but allow it to cool off in air before continuing grinding.

13. Resharpen the cutting edge as soon as it becomes at all blunt. Never keep on working with a blunt tool.

14. Keep strictly to the clearance angles.

15. Adjust the tool so that the cutting edge is exactly central. This very important adjustment should be done with the scribing knife. (However, for external turning of all grades of steel, it is of advantage to fix the tool point slightly above the centre line but at a distance not more than 1 per cent of the diameter of the workpiece).

16. Eliminate all causes of rattling and vibration. See that there is no play in shaft bearings nor in slides. Carbide tools must be used only on machine tools in perfect condition.

CUTTING CONDITIONS.

In principle, carbide tools are intended to take light cuts at high speeds, but some grades of carbide at fairly high speeds will take as deep cuts as high speed steel. Table I. shows results obtained with carbide-tipped tools on a wide range of metals. The machine tools used were of old types, fairly strong and in good condition.

TABLE I.

Material Machined	Type of Tool	Depth of cut in mm.	Feed in mm.	Speed m./min.	Weight of shavings kg./mm.	Life of tool between sharpening in hrs.
Hardened Steel Blank (127 tons, tensile)	1	1.5	0.5	7 to 15	0.060	2
Forged Steel Axle (54 tons)	2	4	0.625	35	0.680	5
Ni-Cr-Steel Container (54 tons), rough cut ..	2	4.5	1.8	21	1.320	2.35
Shell Steel (51 tons), rough cut	2	2.5	0.4	145	1.130	0.75
Forged Steel Shaft (51 tons)	1	3	0.2	149	0.690	1.10
Stainless Steel Tube (55 tons)	1	2.5	0.49	45	0.440	1.25
Ni-Cr Steel Tube (51 tons)	1	4.5	0.33	79	0.925	1.80
Ni-Cr Steel Tube (51 tons)	2	3	1.3	35	1.060	2.50
Cast Iron Drum (180 Brin.)	4	4	0.93	70	1.870	8
Bronze Drum	4	2.5	0.36	190	1.400	6
Bronze Drum	4	2	0.92	144	2.170	4
Ni-Cr Steel Sleeve (57 tons)	1	3	0.27	65	0.410	1.30
Ni-Cr Steel Roller Path (70 tons)	2	2	1.28	19	0.380	0.75
Ni-Cr Steel (54 tons) ..	3	2	1.28	19	0.380	2
Hardened Steel (75 - 120 tons)	2	3	0.75	49	0.860	2.10
Shell Steel, Finish cut (67 tons)	1	5	0.3	16 to 24	0.230	0.45
Semi-hard Cast Iron (230 Brin.)	1	0.8	0.5	109	0.340	3.50
Treated Cast Iron (550 Brin.)	4	3	0.68	43	0.630	4
Special Treated Cast Iron (100 Shore) ..	4	2	0.3	12	0.055	2
Ni-Cr Steel Sleeve, Finish cut (54-57 tons) ..	4	1	0.25	6	0.012	1.10
Ni-Cr Steel (54 tons) ..	1	0.25	0.3	51	0.030	0.90
Cast Steel, Intermittent cut (44 tons)	2	3	0.75	48	0.840	1.70
	3	5	1	25	0.970	0.80

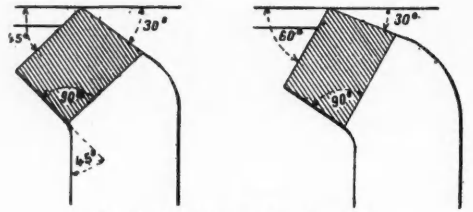


Fig. 1. **Usual types of turning tool.**

TURNING TOOLS.

Fig. 1 shows the 45 deg. and 60 deg. cranked tools best suited to spread the side pressure on the tool shank.

Shock Turning. For interrupted cuts (e.g., out of centre running) the tip should be as large as possible with a large shank to absorb the shocks. Also to protect the tool point a negative clearance should be given (Fig. 2) to allow the shock to be taken behind the point of the cutting edge.

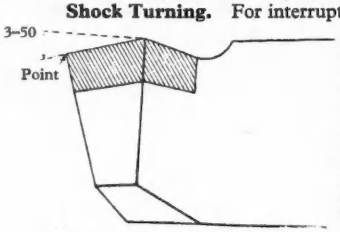


Fig. 2. **Relieved point for shock cutting.**

Tool adjustment. Overhang should be reduced to a minimum (Fig. 3). Special attention must be given to height adjustment.

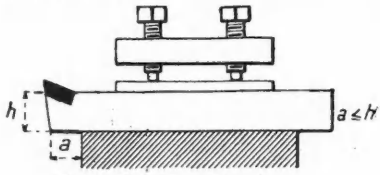


Fig. 3. **Reduction of overhang.**

For turning cast iron and bronze the tool point should be on the centre line (Fig. 4). For all grades of steel, the point should be slightly above the centre line, say :

0.5 mm. for 20 to 100 mm. dia. workpiece
1 mm. for 100 to 200 mm. dia. workpiece
2 mm. for 200 to 300 mm. dia. workpiece
3 mm. for over 300 mm. dia. workpiece.

For boring all metals the point should be strictly on the centre.

Stoppages. Sudden stoppages, owing to motor failure, belt slip or other reasons, generally cause havoc with carbide-tipped tools. If stoppage occurs :

1. Switch off the current or slip off the belt. Do not touch the carriage.

2. Secure the chuck firmly so as to avoid any rotation of the workpiece.

3. Loosen the tool-fixing screws and remove the tool carefully by withdrawing it horizontally.

In this way it may be possible to save the tip which would otherwise most probably be broken.

CONTROL OF TOOLS.

Troubles may occur in service owing to :

(a) Faulty brazing ; (b) Tool grinding being badly done ; (c) Bad conditions of use, e.g., incorrect fitting, excessive overhang, too heavy cut or feed, too great speed, etc. ; (d) Poor condition of machine tool, excessive play in bearings or slide ways, slack belt, undue rattle or even stoppage.

Once tests have been completed with a given tool and the best conditions of service laid down, it should be the duty of the tool-room specialists to see that the operatives are fully cognisant of the best method of using the tool. Regular supervision should be maintained to see that the tool is being used properly and that any troubles are duly tracked down to the causes.

TYPES OF CARBIDE-TIPPED TOOLS.

Carbide-tipped tools have, in general, the same shape and size as high speed steel tipped tools for similar work.

The high price of carbides tends to influence undue reductions in the size of tips but this is not advisable because small tips are more difficult to braze and do not give sufficient mechanical resistance. Small tips are only to be used on light finishing cuts and they do not last long.

For treated cast iron, the line of wear is only slightly inclined from the horizontal and almost all the grinding is done on the top face. It is advisable, therefore, to use a long thin tip as shown in Fig. 5 to allow a number of re-grindings.

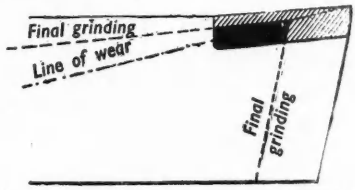


Fig. 5. **Long thin tip for numerous sharpenings.**

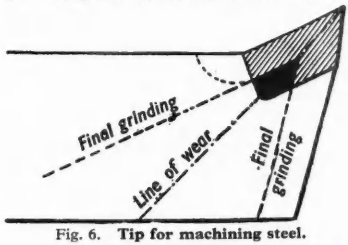


Fig. 6. **Tip for machining steel.**

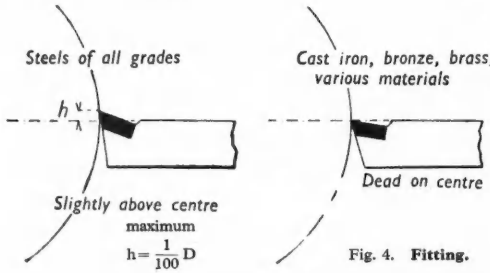
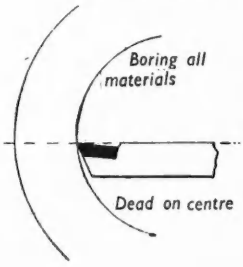


Fig. 4. **Fitting.**



For semi-hard steel, wear of the tip is practically along the diagonal of the cutting angle so that the tip should be shorter and thicker, with a slope equal to that of the line for sharpening, as shown in Fig. 6, to enable full use to be made of the carbide.

Fig. 7 shows the usual types of tipped tools for engineering workshops. The tips, shown on the same scale as the tools,

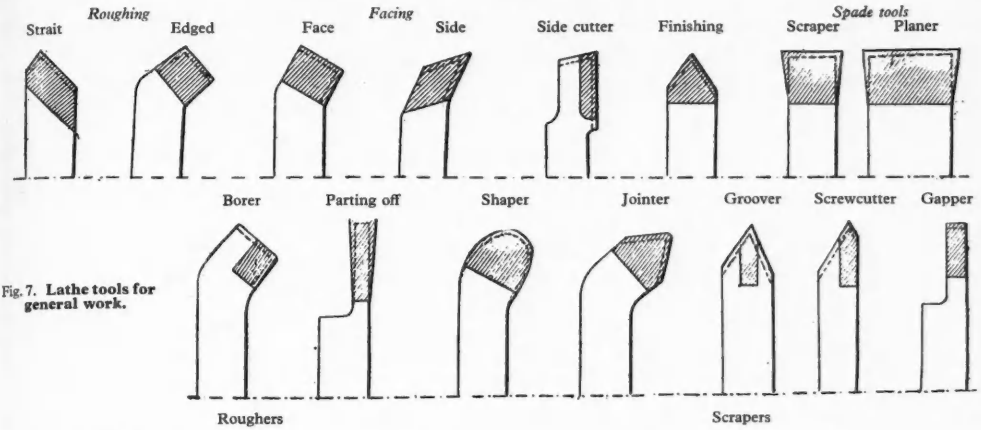


Fig. 7. **Lathe tools for general work.**

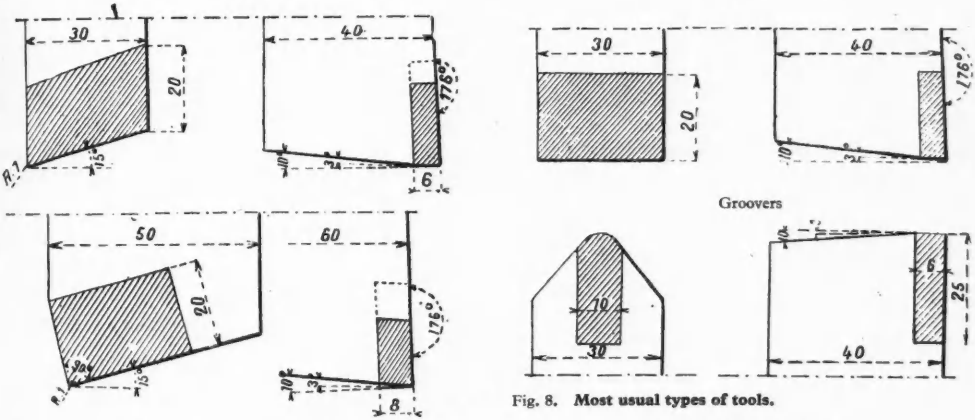
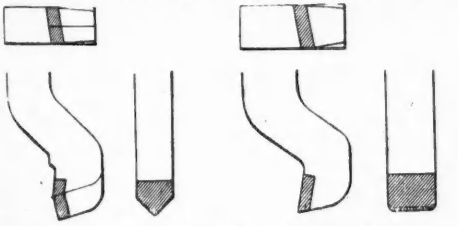


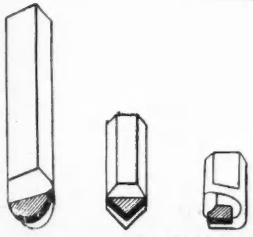
Fig. 8. **Most usual types of tools.**

give a long life if grinding is well looked after. For machining cast iron, the tips shown are particularly recommended.

Fig. 8 shows carbide-tipped tools for planing, shaping, boring and as inserted blades for milling cutters.



Planing and shaping tools.



Boring bits.

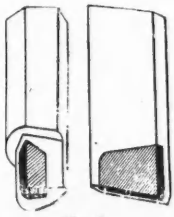
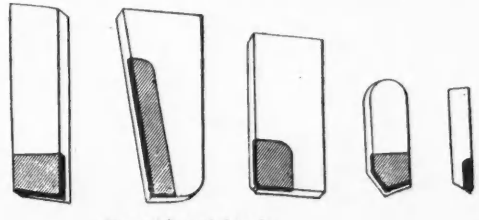


Fig. 9

SPECIAL TOOLS.

Carbide tips can be used for making special tools, mills, drills, borers, etc. Some carbides, very resistant to shock, can be used for planing tools, in which case the tool must have a fitment to prevent the heel of the tip being struck during the return stroke. The planing or shaping machines, using tipped tools, must be built to take them, with high speed, end-blocks, rigid construction and absence of play.

Carbide-tipped tools can be used not only for cutting metal but also for other materials such as glass, porcelain, bakelite, compressed paper, ebonite, synthetic materials, hard rubber electrode carbons. Carbides are also used for horse-shoe gauges, slide bars, guide-rollers, lathe centres, cams, and for the cutting edges of mine drills, trepanning tools, and for



Types of special tools.

wire and tube drawing dies. In these dies, the carbide is not brazed but inserted into the body of the die. Owing to their resistance to heat and oxidation, carbides are of particular interest to the chemical industry.

Tungsten, titanium, and tantalum-tungsten carbides are used for cutting tools. Boron-carbide is harder than tungsten-carbide but is very brittle and cannot be used for drawing dies, although it is used for cutting the tungsten-carbide wire-drawing dies.

For drawing dies, fritted carbides have replaced cast carbides and cobalt, or cobalt-titanium carbides are added to the tungsten carbide, e.g., tungsten carbide plus 6 to 12 per cent cobalt, or tungsten carbide plus 5 per cent titanium carbide plus 6 per cent cobalt.

For insert blades for millers the high price and difficulties of sharpening carbide tips have prevented full development. Special grinding machines are required as grinding takes a long time and means several sets of mills being kept in hand to avoid delays. Consequently carbide tips are only being used on milling tools for very special jobs which cannot be tackled by the best high speed steel mills.

THE PHENOMENON OF METAL FATIGUE.

(From *La Technique Moderne*, Vol. 35, Nos. 23/24, December 1st and 15th, 1943, pp. 189-190).

THE fatigue limit of metals depends upon the composition of the metal itself, its heat treatment, the kind of stress to which it is subjected, the magnitude of the superimposed static stress, the internal stress distribution in the piece, and other important factors.

The way in which fracture proceeds is very characteristic and is regardless of the type of stress prevailing. The fissure begins at a closely defined point, generally in a surface zone where a local stress concentration exists, and grows towards the core of the piece. A study of fatigue fissures is highly instructive, permitting, as it does, the detection of the cause of fracture, and, often, the finding of a remedy. Inspection of the fracture itself makes it possible to ascertain the point of inception and also the direction of stress, as the latter is perpendicular to the surface of fracture. The manner in which the fissure progresses through the material can generally be more clearly seen in the case of service fractures than in those produced under laboratory conditions. This is due to the fact that under operating conditions, oxidation takes place during the periods of rest, while laboratory tests are continuous and of too short duration to allow the occurrence of oxidation. The latter, however, can be artificially produced in laboratory tests by heating the specimen during a short interruption of the test.

However instructive it may be to examine a fracture, it must not be overlooked that the number of stress cycles required for the development, from incipient fissure to final fracture, is only a small fraction of the number of stress cycles withstood by the material prior to the commencement of a fissure. Also, development of a fissure is conditional upon the existence of stress in excess of the fatigue limit at the point of stress concentration. Both the endurance of the material under different stressing conditions and the actual stress concentration prevailing in a given case must therefore be known.

The fatigue limit for alternating stress is based on the cyclic occurrence of stress reversals about zero as mean stress ; or the mean stress may have any tensile or compressive value, if it results from the superimposition of a dynamic and a static stress. The fatigue limit for repeated stress is established on the basis of periodic stress variations between zero stress and maximum stress.

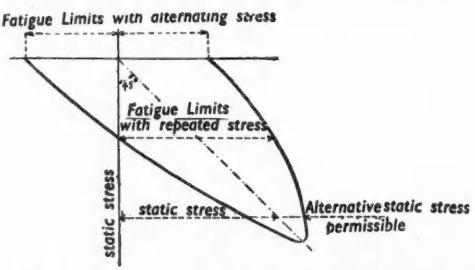


Fig. 1. **Fatigue limits for different types of loading.**

These different fatigue limits can be charted in a diagram of the type exemplified by Fig. 1, where the sum total of variable and static stress is shown as a function of the static stress, the cyclic loading being read off on a straight line encompassing an angle of 45 degrees with the abscissa.

Determination of the fatigue limit is carried out by loading a number of test bars with successively diminished loads until a load value is found at which the test piece endures 100 million stress cycles without failure. Most fatigue limit determinations are carried out by the rotating type of fatigue machine, as this device represents the most simple means of fatigue testing. But in Germany numerous tests have been made with torsional and with tension-compression loading. Unfortunately, the numerous test results extant are not all mutually comparable for several reasons. One of these is the influence exerted by the dimensions of the test piece upon the fatigue strength. Comparative tests have shown that differences in the size of the test pieces may cause the fatigue limit to vary by 10-25 per cent, the smaller piece yielding the higher fatigue limit.

Attempts have been made to establish a relationship between fatigue limit and elastic limit, and it was reasoned that failure cannot occur if the first stress cycle does not produce any deformation. But this has not been confirmed by test results.

The points of stress concentration can be determined and explored by a number of methods, but their determination alone is not enough, as different metals react in different ways. Thus, high tensile steels possessing a high fatigue limit are known to be more sensitive to stress concentration than mild carbon steels, and their advantages are thereby considerably impaired.

An important conclusion to be drawn is that the higher the strength of the material concerned, the more important is the elimination of notch effects. No sharp angles can therefore be tolerated, and the curvatures of the piece should follow the lines of stress. In the case of gears, auto frettage may be employed at the roots of the teeth, thereby producing an initial compression of the material. *Vice versa*, initial stresses in the surface of the piece must be eliminated. Also, surface carbon-depletion must not be allowed to take place in the course of heat treatment, but cementation is considered an advantage. When finishing the surfaces with emery paper, surface scratches must be avoided. In German practice, for reasons such as these, engine reduction gears are ground at the roots besides being well rounded off in these parts regardless of difficulties in manufacture.

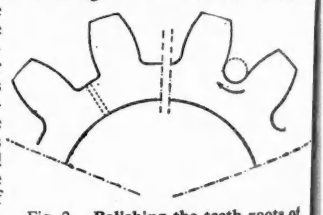


Fig. 2. **Polishing the teeth roots of the reducing gear of a motor.**

HARDNESS TESTING OF HEAT TREATED STEELS.

By O. KUKLA, W. KÜNTSCHER, and K. ROTERMUND. (From *Fertigungstechnik*, Vol. 1944, No. 2, Feb., 1944, pp. 42–44.)

THE difficulties involved in meeting the strict conditions laid down in consumers' specifications with regard to strength of heat treated structural steels have emphasized the need for a method of testing the hardenability of steels. The great importance which attaches to this problem is exemplified by the fact that to produce a 50 mm. thick material with a yield point as high as 90 kg. per sq. mm. and with a notched bar impact strength of 4 mkg. per sq. cm. requires closest adherence to the specified procedure of heat treatment. The interrelationship established between the analysis of the material and the tempering temperature required for this material is charted in Fig. 1. Here it is also seen that a number of heats deviated from the specified compositions as indicated by the curve, and therefore shows abnormal heat treatment requirements.

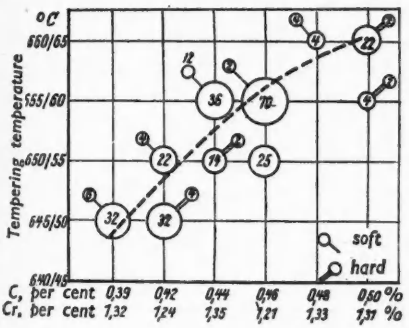


Fig. 1. **Tempering temperature vs. analysis.**

Several test methods have been proposed for measuring the hardenability of steels. The most widely applied consists in preparing several test pieces and measuring their hardness after a complete submersion quench. According to Dr. Krainer's proposal, test pieces in the shape of small plates may be used which are quenched in a hot bath of specified temperature. Of the numerous suggestions which have been published in American publications, the Jominy test has met with special attention. This test consists in water quenching the face of a cylindrical test piece as outlined in Fig. 2. A proposed modification of this test is indicated in Figs. 3 and 4, where a quenching water jet of semi-spherical shape is seen to be used. The size of this jet, which is adjusted by control of the water pressure, provides a measure of the intensity of the quench. After the quench the hardness of the test piece is ascertained on the cylindrical surface at various distances from the face of the piece, the test points being prepared by grinding. In this way hardness distribution curves are obtained as shown in Figs. 5 and 6 for steel St. 60. 61.

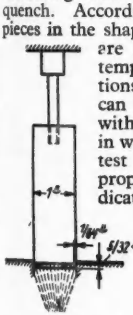
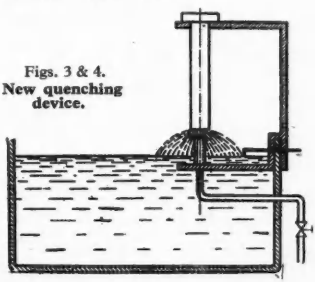


Fig. 2.
Principle of Jominy test.



Figs. 3 & 4.
New quenching device.

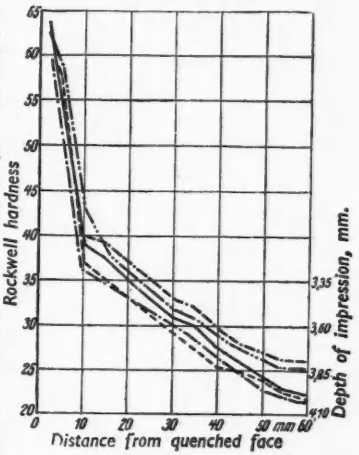


Fig. 5. **Hardness distribution of steel St. C. 60.61.**

These graphs indicate a high degree of surface hardness with relatively small depth of penetration. The influence of the tempering temperature upon the hardness is given in Fig. 6. Corresponding curves of 5 different heats are given in Fig. 7, referring to aircraft steel FLG 1265. In one of these heats the deviation in carbon content is especially large, but the abnormality of its hardening characteristic is still larger than its composition would seem to indicate. The characteristics of eight diffierent heats of aircraft steel FLG 1310 are charted in Fig. 8.

In connection with Fig. 1 it had already been mentioned that it is usual practice to determine the required tempering temperature from the analysis. It therefore rests with the judgment of the responsible party to correctly estimate the influence of the individual alloying constituents upon the hardness characteristics of the material. On an empirical basis, evaluation numbers have been established, the carbon content being given the factor 1, the manganese content the factor 1/5, and the chromium content the factor 1/8. As shown in Fig. 8, the evaluation numbers WZ 74-78 of heats No. 10–16 correspond to hardness values within the areas indi-

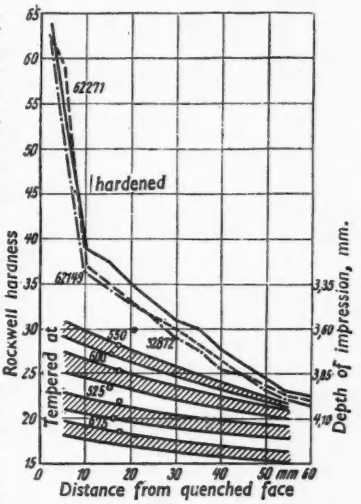
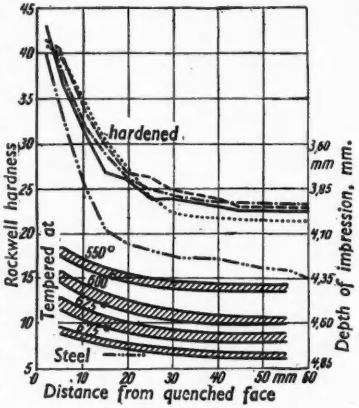


Fig. 6. **Influence of tempering temperature upon hardness and hardness distribution of steel St. C. 60.61.**

cated by shading. Heats Nos. 18 and 19, however, although they have identical evaluation numbers, exhibit markedly abnormal hardness characteristics.

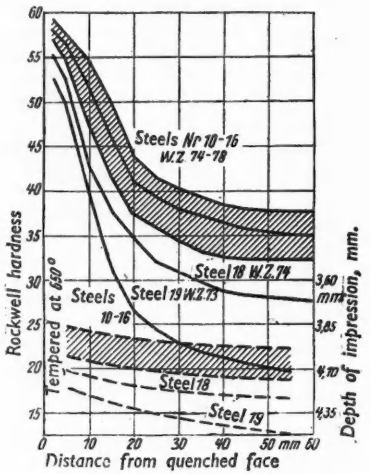
In Fig. 9 three melts of a harder steel with a deeper hardness penetration are shown, the evaluation factors of which are 87 and 88. Of these three heats, steel No. 7 exhibits a pronouncedly abnormal hardening characteristic.

The results of hardening tests conducted with the Cr-alloyed steel TK 12 are charted in Fig. 10. Of the five heats listed, heats 6102, 6108, and 688 were prepared in an acid open-hearth furnace, while heats 1678 and 1778 were made in an electric arc furnace. The different hardening characteristics of these two groups of heats are of particular import as the specifications call for a yield point of 90 kg. per sq. mm. and a notched bar impact strength of 4 mkg. per sq. cm. As evidenced by the chart, current production methods require the employment of different heat treatments according to the method employed in the production of the material.



Steel	C	Mn	Si	P	S
— — —	0.18	2.22	0.44	0.021	0.017%
— — —	0.14	2.30	0.42	0.020	0.026%
—.—.—	0.18	2.22	0.35	0.023	0.019%
.....	0.17	2.26	0.49	0.021	0.011%
—..—..—	0.12	2.00	0.48	0.019	0.016%

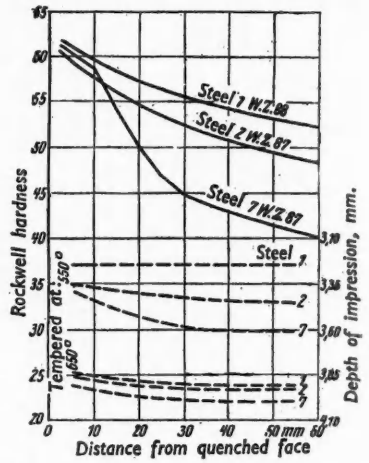
Fig. 7. Hardness and tempering characteristics of five different heats of steel FLG 1265.



Heat No.	C	Mn	Si	Cr	Ni	V
10	0.37	1.76	0.38	0.21	0.11	0.11%
11	0.37	1.80	0.41	0.21	0.09	0.15%
12	0.35	1.81	0.40	0.30	0.08	0.12%
13	0.38	1.70	0.30	0.17	0.10	0.10%
14	0.38	1.78	0.40	0.07	0.09	0.15%
16	0.37	1.73	0.36	0.17	0.06	0.14%
18	0.35	1.64	0.37	0.19	0.24	0.10%
19	0.34	1.64	0.37	0.28	0.16	0.16%

Quenched at 840 deg. C.

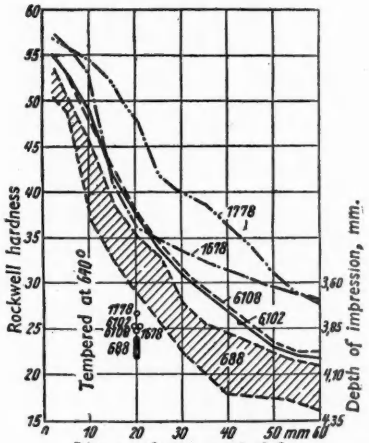
Fig. 8. Hardness and tempering charateristics of eight different heats of steel FLG 1310.



Steel	C	Mn	Si	Cr	Ni	V
1	0.43	1.88	0.39	0.40	0.12	0.15%
2	0.42	1.87	0.43	0.42	0.25	0.17%
7	0.40	2.05	0.75	0.32	0.13	0.15%

Quenched at 840 deg. C.

Fig. 9. Hardening characteristics of three harder steels with evaluation factors 87 and 88.



Steel	C	Mn	Si	P	S	Cr	V
1678	0.46	0.75	0.43	0.022	0.018	1.35	0.13%
1778	0.45	0.65	0.36	0.020	0.013	1.30	0.19%
6102	0.46	0.64	0.53	0.032	0.022	1.05	0.25%
6108	0.41	0.60	0.39	0.036	0.024	1.02	0.22%
688	0.42	0.43	0.36	0.035	0.028	1.10	0.23%

Quenched at 850 deg. C.

Fig. 10. Hardening characteristics of Cr-steel.

TECHNICAL NEWS

Announcements in this section include News relating to British Firms, and on Equipment produced by British Manufacturers.

Available literature may be secured by addressing a request to the advertising department of "The Engineers' Digest," or by writing direct to the manufacturer and mentioning "The Engineers' Digest" as a source.

- ✦ NEW EQUIPMENT
- ✦ BOOKS AND CATALOGUES
- ✦ BUSINESS CHANGES
- ✦ PERSONAL
- ✦ NOTES

PERSONAL

Sir Leslie Boyce, K.B.E., M.P., chairman and managing director of the Gloucester Railway Carriage and Wagon Co., Ltd., has been appointed chairman of Wagon Repairs, Ltd.

Sir Charles Bruce-Gardner has been appointed to the new position of Chief Executive for Industrial Reconversion of British Industry from War to Peace Production, Board of Trade.

Mr. B. G. Churcher, M.Sc., M.I.E.E., has been appointed manager of the research department of Metropolitan-Vickers Electrical Co., Ltd., Trafford Park, Manchester.

Mr. H. W. Healy, M.I.Mech.E., M.I.E.E., Mr. F. Riesenfeld, Ph.D., and **Mr. K. T. L. Sladen, A.C.A.,** have been elected directors of Catalin, Ltd.

Mr. C. B. Kent has resigned his post as sales manager with the Aston Chain and Hook Co., Ltd., manufacturers of non-ferrous metals, Erdington, Birmingham, 24. For the present **Mr. N. I. Bond Williams,** managing director of the firm, has taken charge of the sales department.

Major H. R. Kilner, M.C., has been re-elected President of the Society of British Aircraft Constructors for the year 1944-45. **Mr. A. F. Sidgreaves, O.B.E.,** has been re-elected deputy president; **Mr. H. Burroughs,** vice-president; and **Sir Frederick Handley Page, C.B.E.,** honorary treasurer.

Mr. G. P. Linklater has been appointed works manager of U.D.A. (Plastics) Ltd., Bristol.

Major M. S. McAlpine has been appointed secretary to the British Plastics Federation.

Mr. A. L. McColl has been elected chairman of the Superheater Company, Ltd.

Mr. J. F. Melling has been elected a director of the Projectile and Engineering Co., Ltd.

Mr. F. S. Mitman, C.B.E., chairman and managing director of Messier Aircraft Equipment, Ltd., has been elected a director of the Brush Electrical Engineering Co., Ltd.

Mr. A. Herbert Parkes has been appointed deputy managing director; **Mr. D. W. Douglas,** export director; and **Mr. G. Bowden Parkes,** sales director of Small and Parkes Ltd., Hendham Vale Works, Manchester 9. **Mr. Hugh M. Lawrence,** of the firm's London office at 18, High Street, S.W.19, has been appointed acting London manager of the firm.

Mr. C. A. Powel, manager of the headquarters engineering division of the Westinghouse Electric and Manufacturing Company, has been elected President of the American Institute of Electrical Engineers for 1944-45.

Mr. J. W. Rodger, M.I.E.E., has been appointed managing director of Bruce Peebles & Co., Ltd.

Sir Frank Smith, G.C.B., G.B.E., F. R.S., has been appointed chairman of the Road Research Board of the Department of Scientific and Industrial Research, in succession to the late Sir Clement Hindley.

Mr. N. L. Wright has been appointed Commodore Chief Engineer of Royal Mail Lines, Ltd.

THE INSTITUTION OF MECHANICAL ENGINEERS

Storey's Gate, St. James's Park, London, S.W.1.

FRIDAY, December 15th. General Meeting. Some Considerations in the Design of Class 1 Pressure Vessels, by E. J. Heeley.

FRIDAY, January 26th. General Meeting: Thomas Lowe Gray Lecture, Mechanical Engineering in the Shipyard, by J. Foster Petree, M.I.Mech.E.

FRIDAY, February 16th. Annual General Meeting: The Variable-pitch Marine Propeller, by Dr. J. Lockwood Taylor.

BRITISH STANDARDS INSTITUTION

28, Victoria Street, London, S.W.1.

THE Annual General Meeting of the Institution was held on the 17th October. The Right Hon. Lord Woolton, C.H., P.C., was elected President of the Institution; Sir Percy Ashley, K.B.E., C.B., was elected Vice-President; and Sir William Larke, K.B.E., Chairman of the General Council.

THE BRITISH INSTITUTION OF RADIO ENGINEERS

9, Bedford Square, London, W.C.1.

At the recent Annual General Meeting Mr. G. A. V. Sowter has been elected Chairman of Council for the year 1944/45.

The following members were elected to the General Council:—P. Adorjan, M.I.E.E. (Rediffusion Ltd.); J. W. Ridgeway (Edison Swan Electric Co. Ltd.); H. Brennan, B.Sc. (Universal Relay Ltd.), Gateshead; Lt.-Colonel F. Taylor, B.Sc. (War Office); T. D. Humphreys (A. C. Cossor Ltd.); and M. M. Levy (Standard Telephones and Cables Ltd.). The following members remain on Council for a further year:—G. A. V. Sowter, B.Sc. (Telegraph Construction); Sq. Leader S. R. Chapman, M.Sc.; L. H. Bedford, M.A., B.Sc. (A. C. Cossor Ltd.); N. McLachlan, D.Sc. (Philco Radio Corporation); W. W. Smith, B.Sc. (C.E.B. Birmingham); and J. Dimmick, B.Sc. (Norwood Technical Institute).

TUESDAY, October 31st, Joint Meeting with the Physical Society at the Institution of Structural Engineers, 11, Upper Belgrave Street, London, S.W.1, at 6 p.m. Physics and Radio by Professor E. N. da C. Andrade, F.R.S.

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THE INSTITUTION OF SANITARY ENGINEERS
118, Victoria Street, London, S.W.1.

FRIDAY, November 24th, at 2.30 p.m. at Caxton Hall, Westminster, S.W.1. The Behaviour and Treatment of Iron-bearing Waters, by J. H. T. Griffiths, B.A., A.M.I.Mech.E.

INSTITUTION OF NAVAL ARCHITECTS
10, Upper Belgrave Street, London, S.W.1.

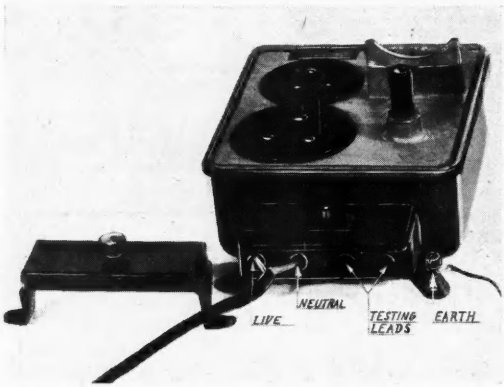
WEDNESDAY, March 21st, 1945. Annual General Meeting.

INSTITUTE OF ECONOMIC ENGINEERING

SUNDAY, November 12th. New Approach to Time Study Initiation, by M. Robson, M.I.Ec.E.

CATALOGUES RECEIVED.

The "Pyrobit" Tester. An illustrated leaflet has just been issued by the Acru Electric Tool Manufacturing Co. Ltd., 123, Hyde Road, Ardwick, Manchester 12. The leaflet describes a new testing device for testing any kind of electric tool and apparatus for continuity; earth connection; leakages, etc. The "Pyrobit" tester is priced at £2/10/-, and the testing spikes, with cables, cost 9/6 per pair.



B.S. 1179-1944. GLOSSARY OF TERMS USED IN THE GAS INDUSTRY.

THE British Standards Institution has issued a comprehensive new Glossary of Terms used in the Gas Industry.

The preparation of this Glossary was undertaken, at the request of the Institution of Gas Engineers, by a Committee fully representative of the Industry.

This publication covers a considerable range of terms in current use in connection with various activities of the Gas Industry, including the production, purification, storage and distribution of gas, and the by-products incidental to its preparation, as well as its utilisation for lighting, heat and power.

Price 3/6 post free.

B.A. BOLTS, SCREWS, NUTS AND WASHERS.

A REVISED edition of the British Standard 57 relating to B.S. bolts and nuts has recently been issued by the British Standards Institution. The previous issue of this specification, which has been in force since 1920, was limited to the dimensions of the heads of the various types of screws and to ordinary nuts. The present edition has been extended in its scope to include requirements defining the dimensions of the complete screws and bolts and nuts. In addition to ordinary nuts, the dimensions are given for lock nuts, and two series of washers are standardised.

Price 2/-.

CONVERSION FACTORS AND TABLES.

A REVISED and considerably enlarged edition of B.S. 350 has been issued, comprising linear, square and cubic measures, measures of capacity, weights, speeds, stresses and pressures, weight per unit length, densities, concentrations, forces, moments, moments of inertia, work, heat, energy and power. There are, also, temperature conversion charts and tables, and wire and sheet metal gauge sizes expressed in decimals of an inch and in millimetres. The book is divided into five parts. Part 1 contains basic tables of units, and definitions affecting conversions. Part 2 consists of conversion factors presented in the form of 27 tables. Part 3 contains multiples from 1 to 3 of 119 conversion factors. In the 32 tables included in part 4 the range is usually from 1-100 units, whilst the 20 tables in common use included in part 5 cover, in general, a range of 1-1000 units.

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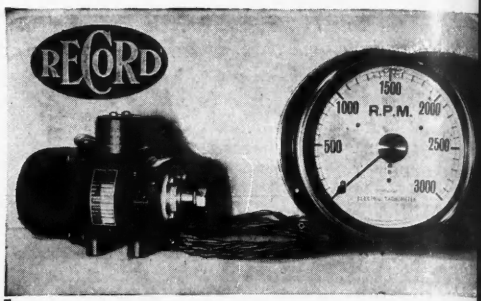

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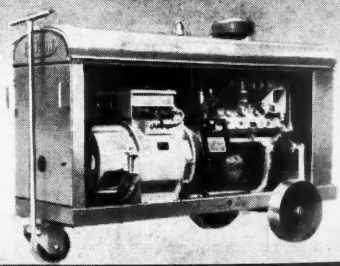

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